U. S. ARMY
TRANSPORTATION RESEARCH COMMAND
FORT EUSTIS, VIRGINIA

TREC TECHNICAL REPORT 61-46

LIQUID METAL REGENERATOR FEASIBILITY STUDY FOR A LIGHTWEIGHT TURBOSHAFT ENGINE

Task 9R38-01-020-08

Contract DA-44-177-TC-687

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prepared by :

CURTISS-WRIGHT CORPORATION
RESEARCH DIVISION
Quehanna, Pennsylvania

HEADQUARTERS
U. S. ARMY TRANSPORTATION RESEARCH COMMAND
Fort Eustis, Virginia

FOREWORD

The work described in this report was accomplished by the Curtiss-Wright Corporation, Research Division, Quehanna, Pennsylvania, for the U. S. Army Transportation Research Command, Fort Eustis, Virginia. This work represents the first step in a research program to determine the optimum regenerative system applicable to aircraft gas turbine engines.

The report has been reviewed by this Command and is considered to be technically sound. The report is published for the exchange of information and the stimulation of ideas.

FOR THE COMMANDER:

APPROVED BY:

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LIQUID METAL REGENERATOR FEASIBILITY STUDY

for

LIGHT-WEIGHT TURBOSHAFT ENGINE

TREC Technical Report 61-46

In partial fulfillment of Contract DA-44-177-TC-687

with

U.S. Army Transportation Research Command
Transportation Corps
Fort Eustis, Virginia

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ACKNOWLEDGEMENTS

The Research Division of the Curtiss-Wright Corporation acknowledges the assistance of certain organizations and individuals in each of their technical fields toward resolving the problems of this project.

The late Professor G. M. Dusinberre of Pennsylvania State University was in charge of the heat transfer aspects of the regenerator designs before his untimely death in December 1960. His contributions are acknowledged with particular feeling.

We thank MSA Research Corporation, Callery, Pennsylvania, our consultants, for the ready application of their specialized knowledge to our problems of handling liquid metal and verifying design concepts: in particular, Dr. R. C. Werner and Messrs. C. H. Staub, R. E. Lee, K. R. Barker and R. Campbell.

The Oak Ridge National Laboratory, Oak Ridge, Tennessee is recognized as the source of a wealth of information concerning liquid metals, much of it stemming from its Aircraft Nuclear Propulsion history. We thank Messrs. A. J. Miller, W. Savage, R. E. McPherson, A. Grindell, W. P. Manly, T. Patriarca, E. E. Hoffman and G. M. Slaughter for making this information available to us.

The Griscom-Russell Company, Massillon, Ohio is fabricating the finned tubes for the cores of the heat exchangers in connection with the testing phase of this program. We acknowledge with thanks the assistance of Mr. R. W. Schroeder, not only concerning the details of the heat exchangers and their cores, but also in the broader aspects of the design and its function.

We are grateful to Mr. G. R. Findlay, President, Liquid-Metals Incorporated, Westford, Massachusetts for his unique contribution in showing feasibility for an attractive flight pump in this application.

ABSTRACT

Feasibility is shown for the use of a liquid-metal-coupled regenerator incorporated in a 7:1-pressure-ratio, 1000-horsepower turboshaft study engine. The weight of the regenerator for .525 overall effectiveness is 235 pounds. The increase in the engine weight resulting from incorporating this regenerator in the powerplant system is 303 pounds. The regenerator design weights are based on current state-of-the-art materials and fabrication techniques for the heat exchanger elements. Use of thin fin material, available before the end of the year would reduce the weight of the heat exchanger to 200 pounds. This fin material development projected two years hence is expected to reduce this weight to 180 pounds. No attempt was made to optimize the engine cycle or powerplant configuration to minimize regenerator weight. The performance of the study engine with regeneration includes the attractive specific fuel consumption at 100 percent power of .446 pounds per horsepower hour, and below .55 pounds per horsepower hour to 40 percent rated power. Detail performance calculations have been made to support the feasibility of this regenerator-engine composite.

A study of the exchangers which transfer heat from the hot gases leaving the turbine to the relatively cooler compressor delivery air has been made. An engine is shown in enough detail to demonstrate how the regenerator system is incorporated by the engine. The cores of the heat exchangers use 5/32-inch Hastelloy B tubing on which .008-inch fins, copper clad with stainless steel, are helically wound 24 to the inch. In the heat exchanger behind the compressor the airflow is in the axial direction through an annulus filled with tubes bent to form involutes. In the heat exchanger downstream of the turbine the gas flow is radial through tubes running fore and aft between the headers. The finned tubes in both cores are serpentine designed in that they take five or more passes between headers, thus closely simulating counterflow heat transfer and also maintaining tube to header welds at a minimum.

The design includes provision for assembly and disassembly of the regenerator system on the engine without disassembly of the engine itself and without the breaking of a NaK line. This provision is accomplished by providing two separate systems, each describing 180 degrees of annulus, each being able to be mounted from each side radially. The headers accordingly consist of 180 degrees of 1-1/2-inch tubing wrapped circumferentially around the engine. Expansion chambers are provided close to the headers and are of similar construction but containing bellows on the inside. The axial cross-over tubes from one exchanger to the other contain bellows to compensate for thermal expansion of the containing system. All components of the regenerator system are free to take positions relative to one another without incurring undue thermal stresses. The material recommended for the NaK containment system is Hastelloy B, which is more than adequate for the containment of NaK at 1100°F for 1000 hours.

The heat exchanger system is designed so that in the event of a failure in the liquid metal system, the spillage of NaK would be internal to the engine and passed off with the exhaust gases of the engine. In the event of such a liquid metal leak in the engine air stream from the heat exchanger system it is expected that the NaK would be consumed. Since the heat release of NaK is small, and since so little is used in the regenerator, it is not expected that leakage of NaK into the engine air system would cause serious damage to the engine.

The interesting range of overall regenerator effectiveness for this design is between .5 and .7. In the interests of coming close to a 300-pound limit for the regenerator system the design was set at a value of .5 for performance study purposes. The actual design as drawn agrees with an overall regenerator effectiveness of .525. The heat transfer minimizes regenerator weight by making the component regenerator effectiveness of the exchanger behind the compressor greater than that of the heat exchanger behind the turbine.

In addition, as presented in Appendix A, a design point study has been made and is presented which describes the functions of fins per inch, overall regenerator effectiveness, and total pressure loss of the air and gas in the exchangers and their effect on exchanger core weight and other physical parameters. A limited study is also included showing the effect of fin thickness and fin material on core weight.

For purposes of this design study NaK alloy (45 Na-55K) was selected as the heat transfer fluid. This selection was based primarily on the fact that this eutectoid remains mushy, under all rates of freezing, down to the freezing point of the eutectic (12°F). Provision is made to thaw the headers and the expansion chambers before engine start-up. A simple electrical system is included in the design and in the weight analysis of the regenerator system. If the ambient temperature is below approximately 50°F, these two components require preheat so that the solid NaK in the core tubes will be able to extrude freely into the headers when quickly heated by the compressor delivery air in one case and by the exhaust gases from the turbine in the other case.

In order to demonstrate the performance of a light-weight, liquid-metal-coupled regenerator for an aircraft turboshaft engine, heat exchangers have been designed for incorporation in this company's Series 300-ST engine. This turboshaft engine is rated at a nominal 60 horsepower and is loaded by an integral centrifugal water pump through gearing. The core design of the heat exchangers, including the headers and the expansion chambers with bellows, closely parallels the design of the 1000-horsepower engine. Minor differences were made between the two designs for purposes of convenience and economy. The basic engine is particularly adaptable to this conversion.

For the testing of this regenerator-engine combination, the NaK loop is designed to be simple and its components are held to a minimum, yet compatible with 20 hours of performance testing. The loop incorporates two electromagnetic NaK pumps in series, an electromagnetic flow meter, an external expansion chamber, provision to pressurize with an inert gas, provision to draw a low vacuum on the system for cleaning and charging purposes, and miscellaneous instrumentation and valves.

CONCLUSIONS

1. A light-weight, liquid-metal-coupled regenerator can be incorporated in a high-performance, light-weight engine in the 1000-horsepower class to successfully compete with other engines for helicopter application. The engine-regenerator powerplant has the following performance and weight characteristics:

    a. Design specific fuel consumption is .446 pounds per horsepower hour; specific fuel consumption is materially below .5 pounds per horsepower hour at the lower usable power levels; as compared to the basic engine at cruise power, this performance shows a fuel saving of 15 percent.

    b. The regenerator system is designed for .3 pounds per engine horsepower output representing achievement of design objective. The weight objectives were met by employing only current state-of-the-art technology except for the flight NaK pump which needs development.

    c. If projected design based on future development were used, a 30-percent reduction of regenerator system weight is possible.

2. Liquid metal (NaK) can be pumped and contained successfully, and with low risk, in this NaK-coupled regenerator.

RECOMMENDATIONS

There are problems in certain areas of the liquid-metal regenerator field for aircraft engines which are most suitable for investigation and require development. Benefits from work itemized below are pertinent to early realization of use of liquid-metal-coupled regenerators in aircraft engines. It is recommended that immediate recognition for future profitable work be given to the following items:

1. Development of an electrodynamic flight pump for liquid metals.

2. Development of a full-scale engine-regenerator turboshaft powerplant for endurance and flight testing.

3. Research to accomplish:

   a. Development of new fabricating techniques including inspection, making possible the use of new materials such as aluminum for fins behind the compressor.

   b. Investigation of liquid-metal composition, the various compositions of Na and K, in regard to their characteristics on freezing and melting and of the possible use of additives such as cesium to NaK.

   c. Demonstration of the degree of fouling of closely-spaced fins on tubes in the air and gas streams of aircraft gas turbine engines.

   d. Development of expansion chamber design.

   e. Demonstration of the assimilation of NaK by the airflow system of a gas turbine engine.

   f. Generation of heat transfer design data for specific optimized designs.

PREFACE

SCOPE AND PURPOSE OF PHASE I OF CONTRACTUAL PROGRAM

As stated in the Recital of the contract, "The typical mission for Army aircraft requires full power for a small percentage of the mission time. It is therefore necessary that Army aviation research include in its program, research projects leading to the development of gas turbine engines which have improved fuel economy when operating at reduced power. The use of regenerators has long been considered as a promising method of achieving part load fuel economy but the weight of the conventional heat exchangers has prohibited their use on aircraft. It is believed that a heat exchanger using liquid metal as a heat transfer medium will be considerably lighter than systems currently used, and will thus be compatible with aircraft requirements". In the Statement of Work of the contract under Phase I the text is quoted as follows in part, "The Contractor shall conduct analytical and experimental studies of a heat exchanger and shall prepare the design of a full-size liquid metal heat exchanger. The mechanical design of the regenerator system will be targeted to demonstrate a weight increase not to exceed .20 to .30 pounds per shaft horsepower at 1000 horsepower, and also based on a regenerator effectiveness of 50% to 70%". The Statement of Work also includes the following instructions, "The Contractor----shall conduct a study and investigation for the design and shall construct a liquid metal heat exchanger suitable for use with a gas turbine engine with 100 h.p. or less and perform tests of the liquid metal heat exchanger installed on a gas turbine engine of known characteristics". The Plan of Performance for Phase I specified use of this company's Series 300-ST turboshaft engine as the test vehicle, its rework to incorporate the required instrumentation and its running to accomplish the calibration of the basic engine.

This report involves, under Phase I, feasibility of the 1000-horsepower engine-regenerator concept and practicality of using the Series 300-ST turboshaft engine for demonstrating regenerator performance. In order that this heat exchanger performance be pertinent to that of the 1000-horsepower design, the design criteria of the heat exchanger cores and headers for the test engine are based on the design study of the 1000-horsepower prototype.

The basic Series 300-ST engine has been calibrated; the calibration report (Reference 4) is being published concurrently under separate cover and herein will be referred to as required.

The work under Phase II and III, the incorporation of the regenerator into the Series 300-ST engine and the performance testing of the engine-regenerator composite, is scheduled for the spring and summer of 1961.

1000 HORSEPOWER ENGINE
COMPREHENSIVE CONCEPT

The degree of success of the design of the 1000-horsepower engine is measured by the featured low specific fuel consumption and in turn the low weight of the engine-regenerator assembly that is demonstrated to be feasible. The levels of these two all important items are more significant on an absolute basis than in terms relative to the basic engine. In other words, compared to the basic engine, the percent of decrease in specific fuel consumption along with the percent increase in weight of the engine regenerator assembly should not be considered controlling. The pertinent question is the performance and weight of this engine with regenerator compared to all other possible engines that might be used for the same application. Previous to the actual design of the 1000-horsepower engine in this program, it was considered that an engine developing 1000 horsepower, weighing less than 600 pounds and achieving specific fuel consumptions, at military power and down to 60% military power, of appreciably less than .5 pounds per horsepower hour could more than compete with any anticipated reciprocating engine or other gas turbine for helicopter application.

Considering the range of Army missions which consist of light observation, surveillance and transport, it is the first two except for ferry missions that the helicopters will perform. In addition to helicopters for light observation and surveillance, VTOL and STOL will have a place but using engines considerably larger than 1000 horsepower and probably totally installed more than 8000 horsepower collectively. The Army helicopter missions are between 2.7 and 3 hours for flight observation and surveillance; for ferry missions, from 10 to 16 hours, the latter for intercontinental ferry range of 2400 nautical miles. A typical cruise power rating for a helicopter engine is 65 percent of maximum power. Altitude is always low enough to be considered as sea level for design purposes. Cruise speed is approximately 100 knots. The fact that cruise requires 65 percent of maximum power is in contrast to lower value of approximately 50% for STOL and 25% for VTOL. Helicopter engines do not have to be operated efficiently at the extremely low power levels required for VTOL and STOL.

Choice of the basic engine required a comprehensive understanding of the purpose of the engine regenerator combination. The fact that specific fuel consumption and engine weight are the two primary considerations cannot be overestimated. The basic engine has to combine low weight and low specific fuel consumption. To establish real minimum specific fuel consumption it is necessary to maintain maximum component efficiencies and minimum total pressure losses in the basic engine. A straight-through axial-flow gas turbine can best meet these requirements. One of the natural advantages that the liquid-metal-coupled regenerator has over the other types is that it can be incorporated with a true axial-flow engine without compromising component efficiencies and without adding unduly to the total pressure losses.

This company has shown in other applications that low specific fuel consumptions at low power outputs can be attained with attractive weight of heat exchangers by placing the heat exchanger associated with the turbine upstream of the power turbine or between power-turbine stages. A configuration of this sort allows the engine to run close to its component design points yet well below the maximum power output of the basic engine. When maximum power is required, the heat exchangers are by-passed or the NaK flow is throttled. What might be considered the more orthodox position for the turbine heat exchanger, that is, downstream of the power turbine, was chosen for the 1000-horsepower engine because, for equal weight of heat exchanger core under conditions of 7-to-1 pressure ratio and 1750°F turbine entry temperature, the location downstream of the power turbine realizes appreciable advantage in specific fuel consumption to that of the upstream location. Figure 1 demonstrates this advantage. The reader will notice in addition that the lower curve associated with the heat exchanger downstream of the power turbine is continuous up to maximum power, giving full advantage of the regeneration over the entire range of powers.

For the 1000-horsepower engine, therefore, the regenerator system is composed of two heat exchangers, one behind the compressor and the other behind the turbine. In the closed liquid-metal system which couples the two heat exchangers NaK is pumped from the heat exchangers behind the turbine to the one behind the compressor thus conserving the heat that would be normally exhausted to the atmosphere. A liquid-metal-coupled system is chosen to recover this exhaust heat since if properly designed, it is inherently compact and light in weight. As pointed out above, the liquid-metal system allows the basic engine to be compromised the least since heat is transferred from the exhaust location to the compressor delivery location with a minimum of rerouting of the gases.

For ease of servicing and to eliminate risk of liquid-metal fires external to the engine, the concept of the engine with regenerator allows for removal of the heat exchangers from the engine without opening a liquid-metal line and provides containment of the entire liquid-metal system within the engine.

Feasibility of the engine regenerator concept is not considered proven on a performance-weight basis alone. The practicality, and with low risk, of containing and pumping the liquid metal has been given full consideration throughout this study.

1000 HORSEPOWER ENGINE

PERFORMANCE

Presented in this part of the study are estimated design and off-design performance characteristics of the engine, considerations leading to the selection of the design point parameter values, and details of the turbo machinery design concept that are closely related to the required performance. The reported performance results were developed through the use of digital computing methods. Existing corporate digital computing programs were used to compute performance with speed and accuracy.

Design Point Performance

To support the choice of the weight flow of the engine and to establish the level of specific fuel consumption at the design point, a design point study was accomplished using the computer. The results of this parametric study are shown in Figure 2. The regenerator effectiveness was chosen at .5 to make the attainment of the 300-pound-maximum goal for the regenerator system most likely. The "Product of Regenerator Gas Total Pressure Recoveries" in the abscissa falls at approximately .94 for the design point, a 2-percent total pressure loss being assigned to each of the cores and to the air diffusion passage between the turbine and the downstream heat exchanger. A 2-percent total pressure loss entering the exchanger behind the compressor had already been taken into consideration as input to the calculations (see Item 12 under "assumptions" in Figure 2). It is noteworthy that the specific fuel consumption at the design point so defined is at the attractive value of .449 pounds per horsepower hour. The basic engine without regenerator has a specific fuel consumption at the design point of .515 pounds per horsepower hour. The following table lists the calculated engine design-point performance including the assigned component characteristics on which the calculations were based:

1000 HORSEPOWER TURBOSHAFT ENGINE

DESIGN POINT PERFORMANCE AND COMPONENT LOSS ASSUMPTIONS

Shaft horsepower, horsepower	1000
Shaft specific fuel consumption, $\frac{lb}{hp\text{-}hour}$	.449
Air weight flow, $\frac{lb}{sec}$	8.25
Compressor total pressure ratio	7/1
Turbine entry temperature, $^{\circ}F$	1750
Turbine adiabatic efficiency, percent	89
Compressor adiabatic efficiency, percent	86.5
Regenerator effectiveness, percent	50
Total pressure recovery of air in core of heat exchanger behind compressor, percent	98
Total pressure recovery of gas in core of heat exchanger behind turbine, percent	98
Total pressure recovery of air entering heat exchanger behind compressor, percent	98
Total pressure recovery of gas entering heat exchanger behind turbine, percent	98
Total pressure recovery in combustion chamber, percent	98
Combustion chamber adiabatic efficiency, percent	99
Total pressure loss at engine inlet, percent	1
Mechanical efficiency (each shaft), percent	99
Total pressure loss in engine exhaust, percent	1
Airflow bleed at compressor exit, percent	3
Power required to drive NaK pump, horsepower	15

The engine as shown above is designed to have a 7-to-1 compressor pressure ratio and a 1750° turbine entry temperature. Choice of these two design values is coupled by the fact that at a pressure ratio of 7-to-1, specific fuel consumption and specific horsepower (an engine weight function) are not seriously compromised by a turbine entry temperature as low as 1750° F (see Figures 3 and 4). To favor low weight, this 1750°F was chosen to be that turbine entry temperature which is usable without resorting to blade cooling. At a turbine entry temperature of 1750°F, specific fuel consumption and specific horsepower are close to optimum when using a compressor pressure ratio of 7-to-1 (see Figures 5 and 6). With weight penalties in mind, 7-to-1 pressure ratio was recognized, in addition, as being the limit within which one stage of turbine to drive the compressor can be employed. A single-spool compressor was chosen because of its mechanical simplicity and light weight. The 7-to-1 compression ratio, in addition, is the limit beyond which variable geometry has to be resorted to in a single-spool engine for good handling qualities.

Turbine

The gas generator turbine is a single-stage unit with a design tip speed of 1293 feet per second, a rotor exit diameter ratio of .78 and a predicited design point adiabatic efficiency of 89%. The free power turbine is also single stage with a design tip speed of 1175 feet per second, a rotor exit diameter ratio of .587, and a predicted design point adiabatic efficiency of 89%. It was necessary to use a single row of turbine exit guide vanes to achieve the high efficiency of the power turbine. A single-stage turbine driving the 7-to-1 pressure ratio compressor is justified by vector diagram analysis. The unconventional shape of the engine and regenerator makes this single-stage configuration possible. A turbine larger than conventional, but in this case consistent with the heat exchanger diameter, permitted attainment of more efficient vector diagrams without incurring stress problems.

Compressor

The compressor is a 7-stage transonic unit with an engine design point pressure ratio of 7-to-1, a predicited adiabatic efficiency of 86.5%, a discharge velocity of 275 feet per second, and an inlet tip speed of 1150 feet per second. Compressor discharge velocity was set at this low level to reduce the diffusion requirement entering the heat exchanger. As in the case of the turbine, the compressor diameter tends to be larger than would normally be found in an unregenerated engine to pass the same air flow. Making the compressor larger to be compatibile with the size of the regenerator has the advantage of placing the resultant design at a high diameter ratio and tends to ease the aerodynamic and mechanical designs. Without resorting to multiple spools or to variable geometry, it is felt that the best way to favor a good stall line is to design the compressor at a slightly higher pressure ratio than the engine design point. A compressor basic design pressure ratio of 7.25-to-1 was selected herein for that purpose. The design feasibility of the compressor was verified by analyzing the vector diagrams at the more critical stages. This investigation proved that the concept of the seven-stage compressor is reasonable.

Off-Design Performance

The off-design performance of the 1000-horsepower engine was calculated in detail on the computer. The first step involved the study of the functions of gas generator and power turbine rpm's as related to specific fuel consumption and shaft horsepower. Engine performance is not sensitive to variations in gas generator rpm, at least down to 70% of its design value and over the range of from 200 to 1000 horsepower. On the other hand, the speed of the compressor spool needs to decrease as the power output of the engine decreases (see Figures 7 and 8). These performance characteristics are compatible with the concept of a free power turbine. Throughout the power range of the engine the rpm of the power turbine is considered to be constant at its design value. The rpm of the compressor spool varies with shaft horsepower as indicated in Figure 8 and in a more conventional plot in Figure 9. Constant rotor rpm is attractive for helicopters over the power range.

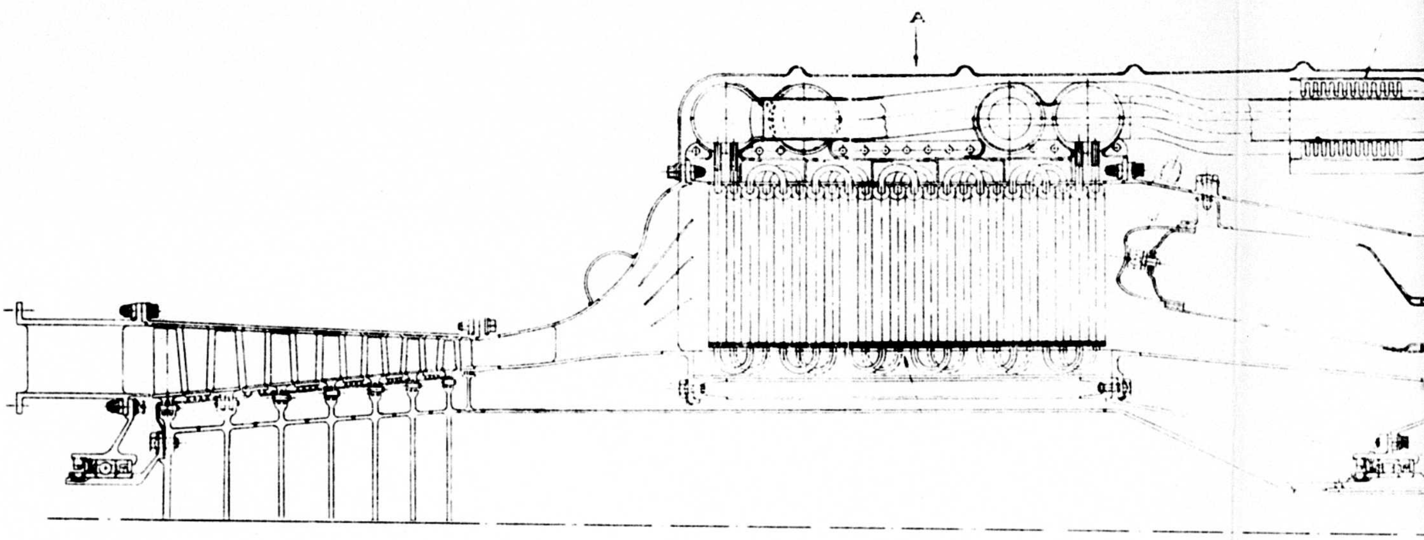
Using the rpm schedules so defined, complete engine off-design performance was calculated under sea level static conditions from 1000 to 300 horsepower. Figure 9 shows off-design values for the two rpm's, turbine entry temperature, specific fuel consumption and exhaust gas temperature. Figure 10 shows off-design performance for fuel flow, air weight flow, compressor total pressure ratio and overall regenerator effectiveness. Figure 11 shows off-design performance for temperatures of the gas and the NaK associated with the heat exchangers and again overall regenerator effectiveness. Figure 12 shows off-design performance for gas velocities, total pressure losses and total pressure levels associated with the heat exchangers. Figure 13 shows the conventional compressor map for the engine and its operating line. Figure 1, already introduced, replots specific fuel consumption as a function of shaft horsepower as compared with that of the basic engine without regeneration (and as compared with another heat exchanger configuration). The engine without regeneration has been adjusted in size, a slight decrease as compared to the engine incorporating the regenerator, so that 1000-horsepower output coincides with engine design point performance. Using an overall regenerator effectiveness (E_o) of .5, the specific fuel consumption at 1000-horsepower output is .449 pounds per horsepower hour and represents a reduction of 12.8% compared to the unregenerated; at 650 horsepower, .484 pounds per horsepower hour, a reduction of 15.5%; at 400 horsepower, .551 pounds per horsepower hour, a reduction of 17.3%. When an E_o of .515 is used, directly applicable to the 1000-horsepower engine regenerator as designed, the specific fuel consumption is .446 pounds per horsepower hour. Note that at the expected cruise power level of a helicopter, 650 horsepower, the specific fuel consumption of the regenerated engine is appreciably less than the unregenerated engine at its design point (6%). With decreasing power the two specific fuel consumption curves diverge as a function of increasing overall regenerator effectiveness (see Figures 1 and 10). This overall regenerator effectiveness curve as applied to engine off-design performance was calculated under the assumption that the ratio of the weight flows times specific heat of the NaK and air remain unchanged* at a value of unity

*This ratio can vary between .95 and 1.2 without appreciable effect on performance (Reference 1, page 18).

at the off-design points as compared to the design point. This requirement presupposes that the pumping of the NaK can be varied independently or that the characteristics of the pump, in this design mounted on the power turbine shaft, can be associated with a small but effective variation in rpm as a function of engine power output. Alternatively, it is reasonable to assume that the NaK pumping rate at constant rpm could be biased by variable geometry in the pump, independently controllable. The best means for maintaining the required air flow-NaK flow relationship at off-design powers would depend on the establishment of the exact pump characteristics (refer to Page 21 for a description of the NaK pump concept).

The engine off-design characteristics are reasonable. Stable engine operation throughout the power range is indicated by the position of the operating line on the compressor map (see Figure 13). In Figure 12 the differing characteristics of the percent total pressure loss entering the cores and in the cores of the exchanger behind the compressor and the exchanger behind the turbine are worth comment. The loss behind the compressor on a percentage basis increases slightly with decreasing horsepower of the engine, while that behind the turbine decreases to half its original value. Note that the effect is due not only to a decreasing velocity entering the exchanger behind the turbine but to the fact that the pressure level behind the compressor falls off rapidly with decreasing engine output while the pressure level leaving the turbine remains constant. These pressure levels are those on which the percentage loss is based, engine performance in turn being sensitive to the percentage loss.

The indicated off-design engine performance estimates represent the results of an investigation which accounts for the off-design performance variation of all major components. Compressor and turbine maps were used to represent typical off-design characteristics for units of the subject types. Off-design performance of the regenerator system was estimated and was included in the resultant performance analysis. By recognizing that many component variables exist and by taking realistic account of them in this performance analysis, it is felt that the indicated results present reasonable and attainable performance.



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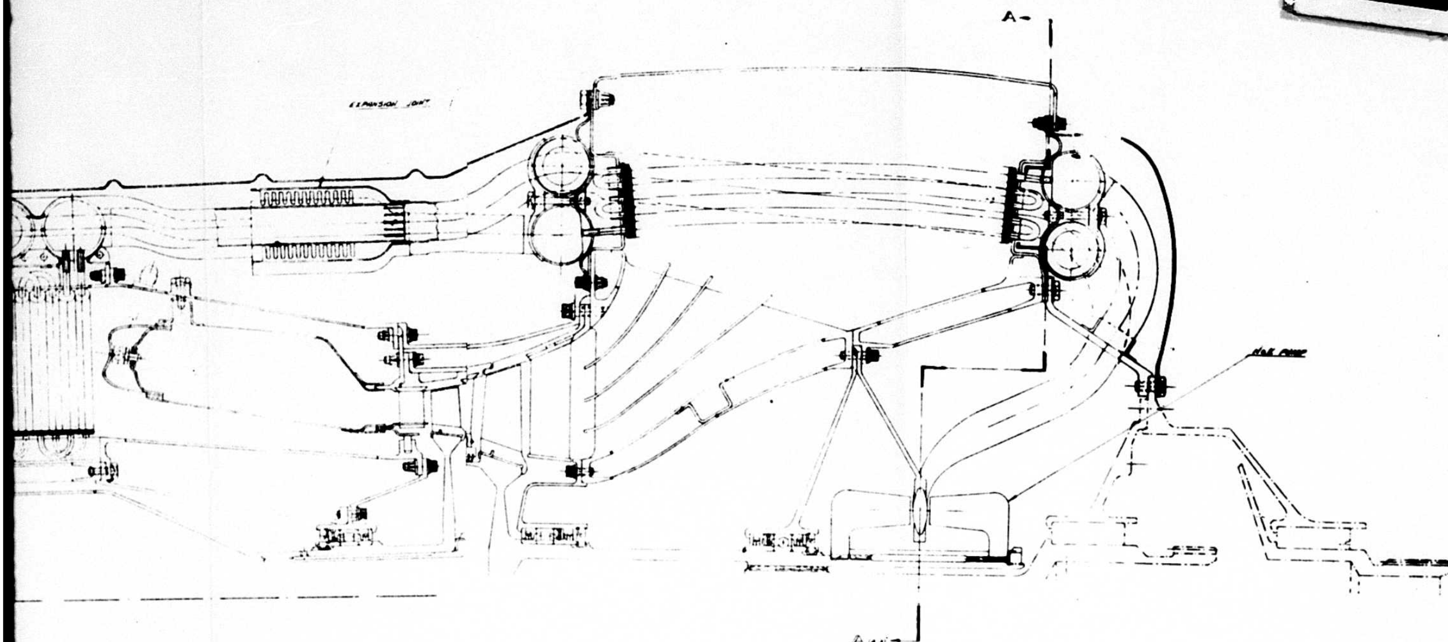

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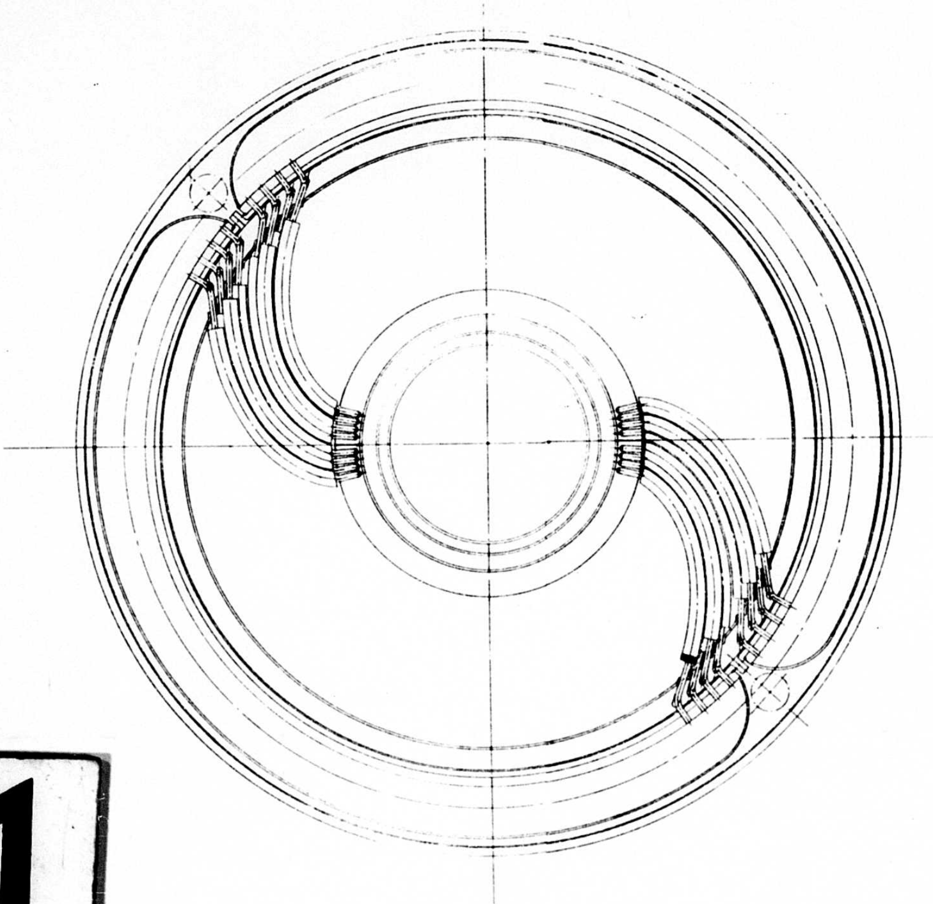

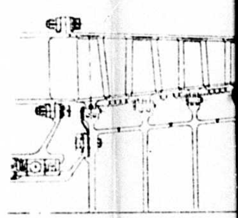

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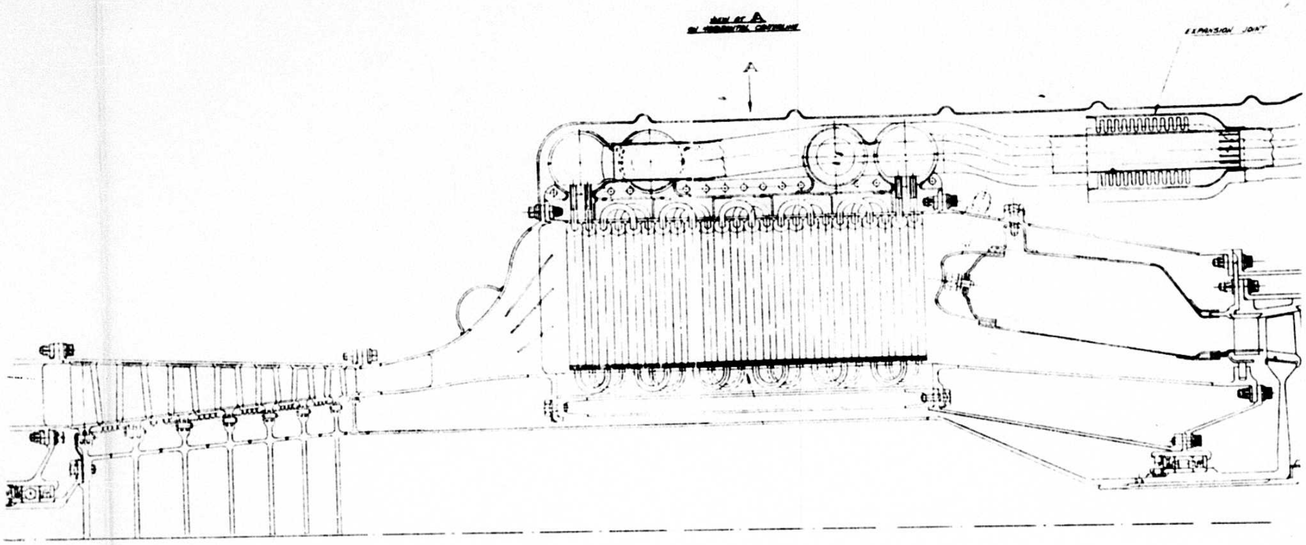

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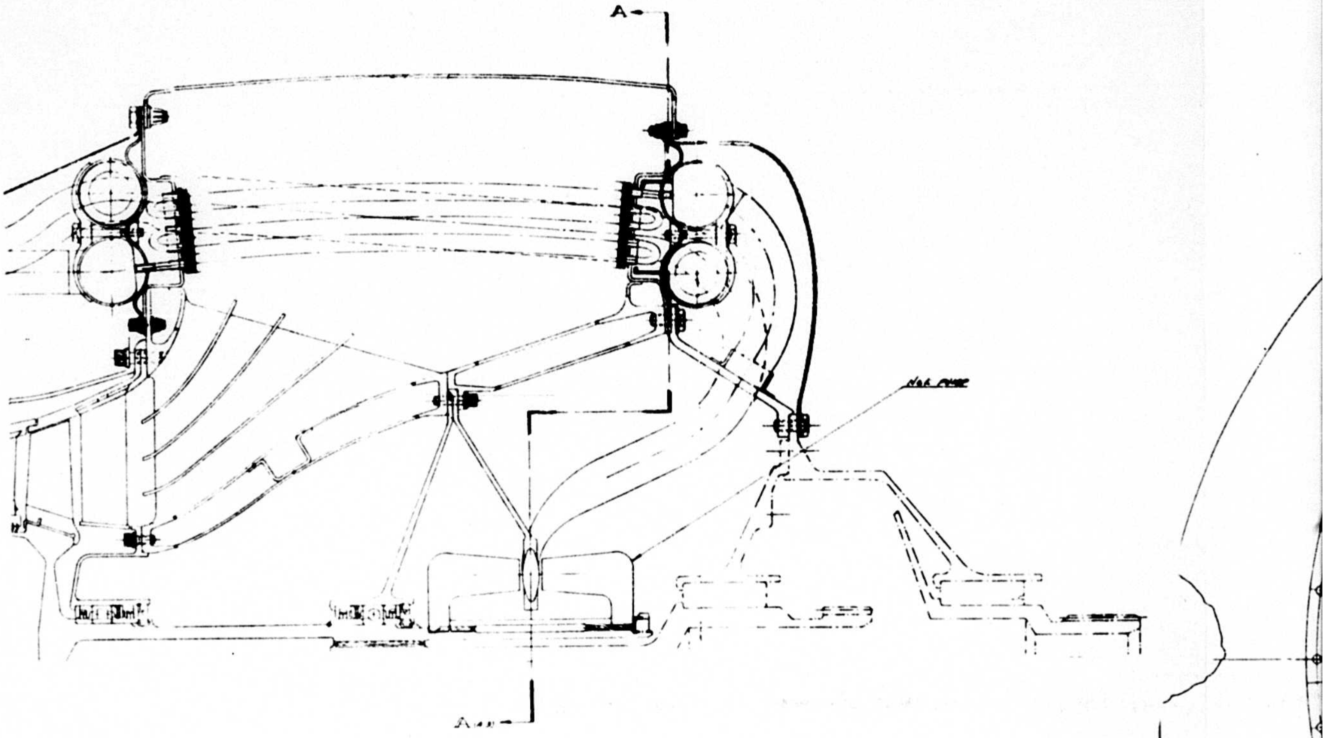

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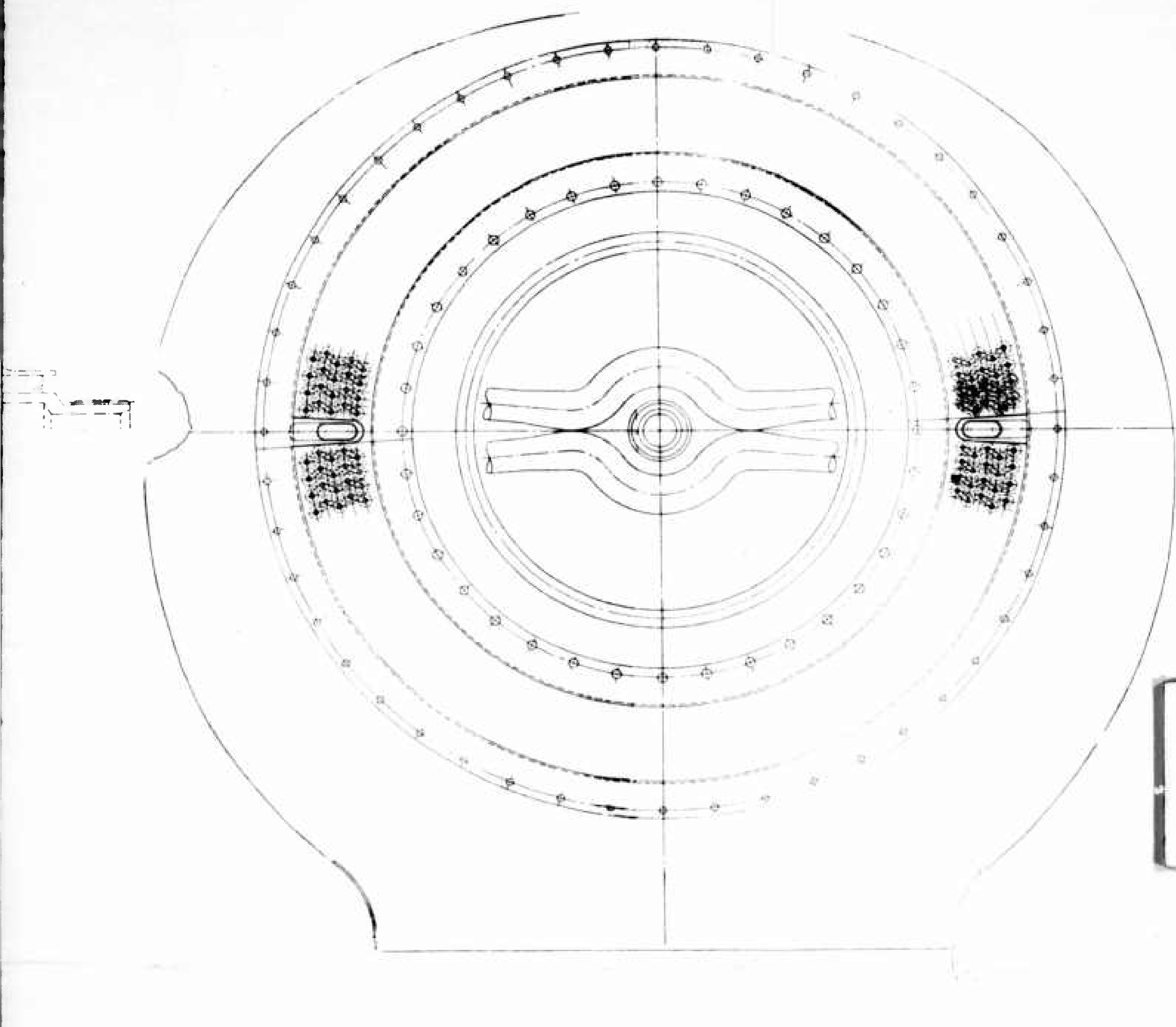

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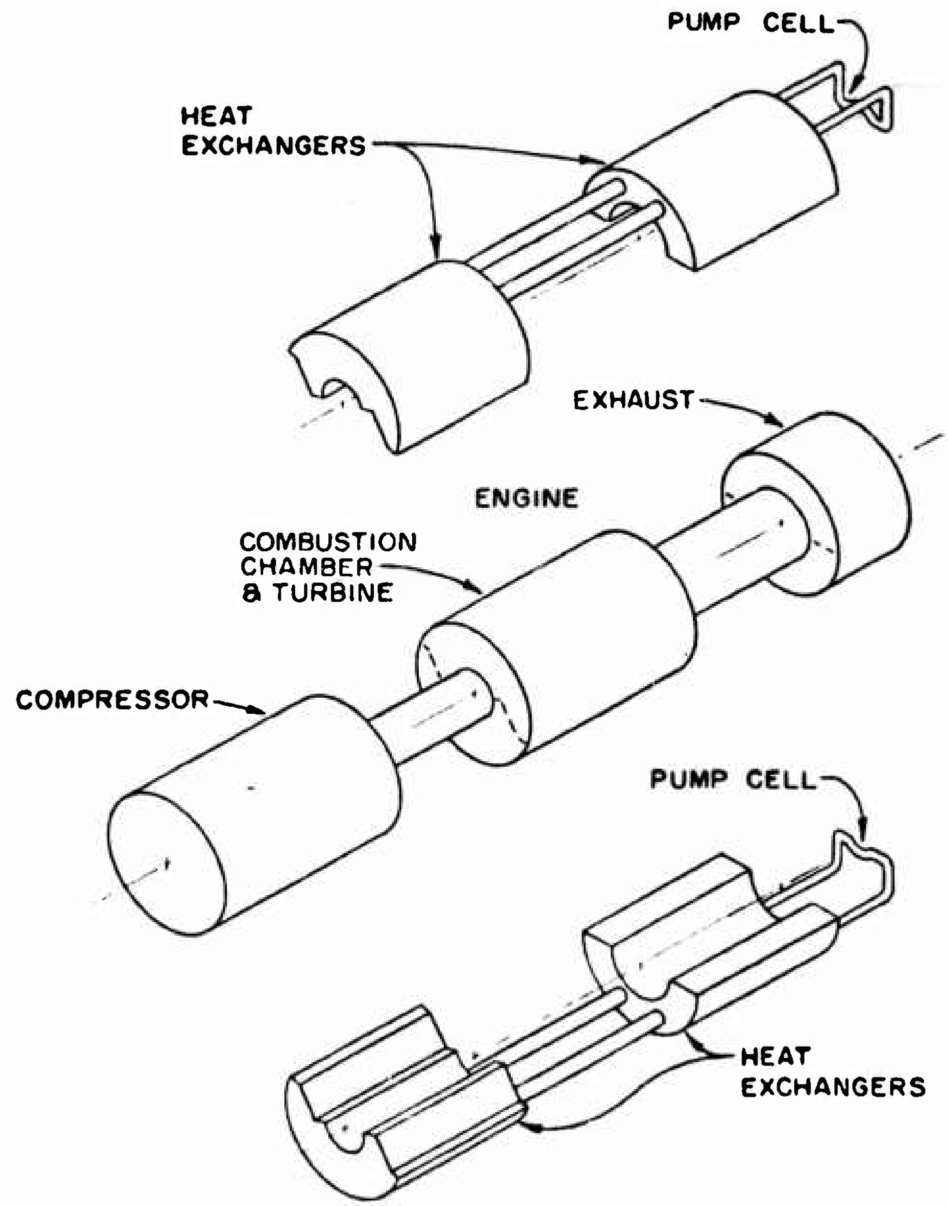
R 15347 (3)

SCHEMATIC DIAGRAM SHOWING ASSEMBLY AND REMOVAL OF REGENERATOR UNITS WITHOUT OPENING A NaK LINE.



1000 HORSEPOWER ENGINE

MECHANICAL DESIGN OF HEAT EXCHANGERS

Drawing R15347 (pages 13 - 15) shows the layout study of the heat exchangers and the regenerator system of the 1000-horsepower engine.

The general configuration of the heat exchangers is mainly established by the thermodynamic considerations such as the required frontal area, the number of liquid-metal passes and the number of layers* of tubes. In this design the heat exchanger behind the turbine has four times the frontal area and about one-fifth the number of tube layers as the compressor exchanger. When an attempt is made to design these heat exchangers into the air passage system of the engine, the above requirement dictates that the heat exchanger behind the compressor have considerable length with the air flowing through in an axial direction and the exchanger behind the turbine be radially thin with the gas flowing through from inside to outside.

Since the heat exchangers are large components of the engine, a problem arises as to the assembly and disassembly of the units without compromising the engine design excessively and without disconnecting the liquid metal lines. The exchangers are annular in shape, yet in two separate units. The annuli are split in an axial sense and are formed when the two integral parts are assembled radially from top and bottom (see schematic diagram on page 16). A separate pump cell is provided for each system. To remove the regenerator system it is not necessary to disassemble any part of the engine, with the possible exception of the reduction gear box. This feature is particularly pertinent since the present state of the art of the containment of liquid metals for other than laboratory purposes prescribe that the flow circuit have all welded connections; hence, no mechanical joints. Each unit of the exchanger behind the turbine is permanently connected by cross-over pipes to the respective unit of the exchanger behind the compressor.

The structural design and weight optimization of the liquid-metal containing system is dependent upon a compromise between the passage flow length of the two fluids, air and liquid metal. To accomplish efficient heat transfer, an exchanger requires a multiplicity of passes of the liquid metal; i.e., to simulate counterflow. A large number of passes imposes a serious pressure-drop parameter on the containment system, thereby increasing pump size and piping wall thickness. The compromise must be made between the number of passes and the system pressure. From heat transfer analysis data the difference between the exchanger effectiveness with four passes and an infinite number of passes is small (less than 5%). Therefore, a design criteria can be established in order to keep the pressure as low as possible; the desirable number of flow passes is four. From a heat transfer standpoint the frontal area and number of tube layers

* "Layers" of tubes in this report refer to planes of tubes across the air or gas stream. Progression from one layer to the next is in the direction of the air or gas flow.

are established. From a mechanical design standpoint it is desirable to have as large a flow area as possible, therefore, low velocities. A reasonable maximum liquid-metal velocity in any part of the system is 30 feet per second. Average NaK velocity in the serpentine tubes of the exchanger behind the compressor is 13.4 feet per second and behind the turbine, 15.5 feet per second; in the headers, 7.75 feet per second and in the cross-over tubes, 17.4 feet per second. These velocities together with entering and exit losses incur a total pressure drop of 40 pounds per square inch.

The mechanical configuration of a liquid metal regenerator system for flight vehicle powerplant should be established in close congruity with the other components in order to maintain compactness. Compactness is obtained principally by designing for the efficient transfer of heat from metal to air in the smallest volume.

Finned Tubes After considerable study in the nuclear field of heat exchangers this company is convinced that small tubes with closely-spaced circular fins result in minimum size. The exchanger cores incorporate serpentine tubing thus keeping the headers, in turn, to minimum size and reducing the number of welds joining tubes to headers. These finned tubes are fabricated in this design from .156-inch outside-diameter tubing with a .015-inch wall thickness. Fins are helically wound and brazed to the tubing. The outside diameter of the fins is .418 inches; fin thickness is .008 inches, consisting of .004 inches of oxygen free high-conductivity copper and .002 inches of AMS 5521 stainless steel cladding on each side. The fins are spaced on the tubes 24 to the inch. Tubes are spaced to each other at .458-inch intervals in an equilateral pattern, leaving a .040-inch clearance between fins. Weight per inch of this fin tube is .00903 pounds. Although a saving of weight is indicated in the heat transfer study in Appendix A by designs using 30 fins to the inch, 24 fins to the inch are used in this design to avoid fouling of closely-spaced fins in the air stream and particularly in the hot gases downstream of the turbine.

Since mechanical fatigue is possible where air is moved at high velocity, material selected for those elements in the air stream favors high fatigue strength in the operating temperature range as well as compatibility with the liquid metal and hot gas. Hastelloy B was selected as the tubing material, and for the other components of the NaK containing system, because of its properties at 1100°F. These properties include good resistance to corrosion by NaK and by air, high 1000-hour rupture and creep strengths and, in particular, high fatigue strength. Other materials with good fatigue strength which are recommended for tubing to successfully contain NaK at these temperatures are HS 25 and Hastelloy N. Mass transfer in 1000 hours at 1100°F maximum is negligible with these materials.

Liquid NaK will quickly attack local areas where inclusions or defects in the material are concentrated. It is even conceivable in thin wall tubing that a minute oxide particle will cause an early failure of an otherwise acceptable heat exchanger. In addition very small amounts of impurities found in the liquid NaK itself can greatly accelerate corrosive attack. For these reasons and to improve fatigue strength, both the heat exchanger material and the NaK must be reactor grade and free from impurities.

It is equally important that all weldments exposed to the liquid NaK be resistant to corrosion by the liquid. The presence of even a small amount of oxides or gas entrapment can result in serious localized corrosion. Therefore it is imperative that the material be readily weldable without danger of weld defects, undesirable phase changes, or loss of ductility and other mechanical properties. These conditions are met by the materials recommended.

Heat Exchanger Behind Compressor

Integrity is maintained only through careful design of each component to operate in its environment. Due to the high thermal gradients, the effect of expansion on each component should be considered and the resulting strain eliminated where possible.

The attachment of the tubes to the headers represent a considerable metal section change; therefore, the welding and back-brazing technique if used, should provide generous tube to header fillets. The preferred technique is to expand the tube in the header holes and weld only. To minimize the number of tube-to-header welds, the tubes should be serpentined, i.e., folded across the air flow passage as many times as practicable.

An involute design is merged with the serpentine tube concept in the core of the exchanger behind the compressor. This design in addition to its structural considerations provides maximum utilization of frontal area for the axial flow of air and with constant NaK velocities at any radius. The curvature of the tubes describing the involute provides stiffening in one plane and thereby increases their natural frequency. The calculated first bending mode frequency of the straight fin tube was found to be 18,700 cycles per minute. Bending stresses imposed on these tubes are small. Calculations indicate a maximum stress of 296 pounds per square inch when the tubes are considered cantilevered. Actually the support system for the tubes approaches a beam fixed on one end simply supported on the other, implying even lower stresses. All tubes are mounted so that they can expand without restraint thereby eliminating any longitudinal stresses due to thermal growth.

An entrance header and an exit header only are required with serpentine tube design. Hastelloy B is specified for the headers, as for the tubes as mentioned above; one-and-one-half-inch inside-diameter tubing with .045-inch wall thickness is used. The header cross-section is circular in shape with a heavier wall where the exchanger tubes enter, thus allowing a minimum weight. The 180-degree headers are made in two halves to permit welding of the tubes along their long axis. Support of the finned tubing on the outside diameter is accomplished by punching and stamping lipped holes in sheet-metal shells. The inner end of the tubes is supported by a fabricated radially-slotted cylinder. All of the gas pressure loads on the outer diameter of the heat exchanger are taken up by the second outer shell which has been designed to take a calculated stress of 21,100 pounds per square inch. All of the supports and other components in the 180-degree segments of the exchanger behind the compressor are designed to be as light as possible not only to minimize weight but to allow flexibility in reducing thermal stresses. The ends of the 180-degree headers in each segment are mechanically attached to each other so that thermal growth results in increased diameter but not straightening. Header circumferential stress due to an operating NaK pressure of 50 pounds per square inch is 894 pounds per square inch.

Heat Exchanger Behind Turbine

The heat exchanger behind the turbine is designed to accept radial gas flow and has the same serpentine finned tubing as in the other exchanger, but running fore and aft. This tubing in addition is bowed radially outward, that is, in the direction of the air flow to promote stability and lower stress. In this exchanger the finned tubes are supported in plates with punched holes and sleeve inserts for bearing surface. These tubes are also free to expand to take care of thermal growth. Attachment of the headers to the supporting structure is by mechanical clamping to the two contour plates which allows circumferential expansion. Bending stresses in the finned tubes are slightly higher than in the exchanger behind the compressor due to the greater span.

Piping and Bulkheads

To maintain a light structure the bulkheads and piping should conform as near as possible to good pressure-vessel design. This criterion calls for the elimination as far as possible of flat surfaces and the use of generous radii. Connecting joints between bulkheads and inlet and outlet piping must have good flow transition. Cross-over pipes between the two exchangers must provide for thermal expansion. The most common device used for thermal expansion absorption is a pipe loop. When space is of prime consideration, the bellows joint is more suitable.

Connecting lines between headers are one inch in diameter with .045-inch wall thickness, and fabricated of Hastelloy B. These connecting lines are attached to the headers by welding to flanged holes, formed with large radii in the headers. In order to compensate for thermal differentials between the upstream and downstream exchangers, bellows units are built into the connecting lines.

Expansion Chambers

Contraction of the liquid metal during cooling and expansion during heating must be taken into account by the provision of sufficient expansion volume. Need for expansion chambers is well illustrated by the fact that NaK (45Na-55K) expands 20% in volume for a 1100° F rise in temperature (100°F to 1200°F). Since the expansion chamber must operate at any attitude, the free-surface type with an inert cover gas is unsuitable for this application. Bellows have been incorporated into the expansion chambers in this design. NaK is contained on the outside of the bellows against the pressure of an inert gas on the inside. The expansion chambers are shaped like the headers using the same one-and-one-half-inch inside-diameter tubing with .045-inch wall thickness. The NaK is on the outside of the bellows to make it possible to properly drain. Each expansion chamber is located close to its header section and is connected to each by half-inch tubing. The bellows in each expansion chamber is designed to allow for 27.5 cubic inches displacement; in the four expansion chambers, 110 cubic inches total displacement. The volume of the entire NaK system without expansion chamber is 687.5 cubic inches. Between the melting point of the NaK and the maximum operating temperature this volumetric increase is considered adequate. Expansion of the containing system itself is not estimated quantitatively but is considered to be a real factor of safety.

Liquid Metal System Containment

The liquid metal system is completely contained inside the engine. A one-piece titanium alloy shroud .030-inches thick slips over the heat exchanger behind the compressor and the connecting lines and is bolted to the supports of each exchanger fore and aft. If a leak should occur in any part of the NaK system, a local and internal fire would break out and burn at the source of the leak if the NaK were at its operating temperature. Progressive damage to the hardware at the leak location and ensuing enlargement of the leak would take place. The heating values of sodium and potasium, however, are small, 3891 and 1984 BTU per pound respectively. The titanium shroud is designed to contain both the products of the NaK combustion and the heat. This shrouded internal volume is vented into the exhaust system of the engine. A white smoke in the engine exhaust would be indication that a NaK leak had occurred. Damage to the heat exchanger is expected to be restricted to the localized area of the leak; the engine is expected to carry off the products of the combustion and the heat without damage to itself. The products of combustion NaK are irritating but not toxic.

NaK Pump

A liquid-metal pump for operation in an aircraft should combine the following desirable features: lightweight, compactness, adaptability to the engine drive, leak-free operation at any attitude and reasonable life. Mechanical liquid-metal pumps and the electromagnetic liquid-metal pumps are currently available for stationary use. The mechanical pump generally employs a centrifugal impeller to move the liquid metal.

One critical component of this type of pump is the seal between the liquid metal and air or other fluids. The seal has to prevent any contact of the liquid metal with air or with hydrocarbons. The other critical component consists of the bearings which must operate at high temperatures and, for some designs, in contact with the liquid metal.

Mechanical pump seals developed to date are classified as canned-rotor, cover-gas, and frozen seals. One type of canned-rotor or membrane-sealed pump employs a thin non-magnetic membrane that is permanently sealed between the liquid metal and air. An external rotating magnetic field coupled through the membrane to a rotor which is submerged in the liquid metal provides the driving torque. The obvious advantage of this type of pump is the zero-leakage feature regardless of attitude. The gas-seal pump operates on the principal of using an inert gas chamber above the free liquid-metal surface. This design is unsatisfactory for aircraft application due to the movement of the free surface with change in attitude. The frozen seal pump, as the name implies, accomplishes sealing by freezing the liquid metal in an annulus between a stationary and a rotating element. While this pump can seal with zero leakage, an undesirable feature is the requirement of a refrigerating system for the success of the liquid-metal containment.

The electromagnetic type pumps move the liquid metal through a continuous tube loop by taking advantage of the electrical conductivity of the liquid metal. In the Faraday type, basically DC, the pumping force results from an external current flowing in the liquid metal during its passage through a magnetic field. The Faraday types, both AC and DC as developed to date, are impractical for flight installations due to excessive size and weight and the requirement of large amounts of external electrical power.

Certain pumps called "electrodynamic" are designed on the induction principle. In this type a moving magnetic field induces current in the liquid metal. The reaction between the eddy currents so induced and the moving field results in a pumping force on the fluid. This electrodynamic pump is the only pump in the small size required by this project which appears feasible and adaptable to flight engine application. Its compactness, lack of seals and adaptability to the engine installation are together uniquely attractive.

The pump shown in this design is patented. The current state of development of this type of pump permits accurate estimates of performance and weight. The pump consists of two permanent magnets mounted on the main shaft of the engine. Inserted between the magnets near the periphery are two 180-degree "cells" of the NaK piping. Each cell is incorporated in, and pumps NaK through, the closed circuit of each system. The 180-degree cell of the pump piping can be pulled away radially from the pump magnets with the heat exchangers at disassembly and without breaking a NaK line.

Each pump cell has a capacity of 40 gallons per minute at a developed output pressure of 45 pounds per square inch. The pump weight is approximately 30 pounds and requires 15 horsepower to drive. Quoting from a letter by Mr. Gordon R. Findlay, (see acknowledgements page), to Mr. Hargus Watts of this company dated during the latter part of October 1960, there follows an account of the pump development required: "In order to realize and optimize this performance, it would be necessary to carry out an application development program aimed at suitably modifying the standard configuration of our electrodynamic pump to meet the specific application proposed. We estimate that the program would require about 9 months and cost in the vicinity of $60,000. The program would comprise an initial application engineering design study in which magnetic circuit, electric eddy circuit, and hydraulic circuit parameters would be selected to give an optimum weight and performance package. This engineering phase would in turn lead to an actual flight pump design. Finally, a test pump and loop simulating actual shaft speed and loop conditions would be set up and operating data obtained to verify the design".

NaK Heating System

A heater system has been incorporated into the exchanger headers, expansion tanks and connecting lines to adequately melt the NaK before engine start up when ambient temperatures are less than approximately 50°F.* An electrical system was chosen because of its simplicity and light weight. This system uses aircraft power supply for operation. Basically this NaK heating system consists of rod elements through the headers, easily replaceable, and bonded surface heating elements on the expansion tanks and connecting lines. Power requirements for this heater have been established at approximately 1/3 the output of a 24-volt aircraft battery with a 75 amp-hour rating to accomplish unfreezing of the headers and connecting lines from the low temperature of -65°F. Starting procedure and preheating of the expansion tanks and connecting lines are involved in the choice of the exact NaK composition chosen for this design. Further discussion of these subjects is found in the section which follows.

Liquid Metal Composition

The possibility of using one of two high-temperature-melting-point liquid metals, lithium and sodium, as well as one very low-temperature-melting-point eutectic, cesium-potassium, was considered. Lithium and sodium are attractive for their higher specific heats and electrical conductivity, and the cesium-potassium eutectic is attractive for its -60°F freezing point, which would eliminate freezing and consequently any provision to thaw. In comparing specific heats the advantage of sodium over NaK (56Na-44K) is a modest one: .307 compared to .248 BTU per pound per degree F. These specific heats are an inverse measure of liquid-metal flow requirements on a comparative basis and in turn a saving in weight of not only liquid metal but of the containing hardware.

* Thawing of the headers and expansion chambers before engine start up is predicated on the concept that solid extrusion of NaK from the finned tubes into the headers has to be accomplished to protect the tubes during start up. It is possible that voids in the solid NaK and the flow of mushy NaK may adequately provide volume for expansion without preheating. In this sense the requirement to preheat is based on the most severe possibilities.

The electrical resistivity of sodium is approximately 1/3, and that of lithium approximately 1/2, that of NaK. The pump manufacturer advises that the use of sodium instead of NaK in the system, because of this electrical advantage, would reduce the weight of the pump from 30 to 20 pounds. Use of the cesium-potassium eutectic implies a specific heat of less than half that of NaK and an electrical resistivity perhaps twice that of NaK. The performance disadvantages of the cesium-potassium eutectic are appreciable.

The choice of NaK as the liquid metal in the heat transfer system of this design is predicated on the concept that NaK in the tubes, the headers and the connecting lines can be frozen after engine shut down and thawed successfully before engine start up in routine fashion. Use of NaK (45Na-55K) requires heating to 45°F to completely thaw. Use of lithium and sodium would require heating to 357 and 208°F respectively. The concept of a successful start up which requires preheating of the headers, expansion chambers and connecting lines reasonably presupposes that solid NaK in the finned tubes of the core, when suddenly heated by the compressor delivery air in the one exchanger and by the turbine exit air in the other exchanger, will extrude into the headers without damaging the tubes. Accordingly it is necessary that the headers be free and the expansion chambers operable. Preheating of the connecting lines is prescribed to avoid delay in NaK circulation and regenerator operation.

To successfully thaw after freezing it is advisable further to reverse the sequence of the freezing events, so that the solid NaK which is the last to freeze will be the first to be melted and thereby have some local voids into which to expand. This concept leads to a refinement in the choice of the NaK composition. It can be shown on the sodium-potassium liquidus diagram that the conventional NaK (Na56-K44) may, depending on the rate of cooling, precipitate sodium out of the composition and thus locally form a complete solid before the freezing temperature of the eutectic (12°F) has been reached. Use of the NaK composition (45Na-55K) is specified because it assures a mushiness of the NaK to be present down to the freezing point of the eutectic. Below this freezing point a porous structure of the frozen NaK is expected. The temperature of initial freezing of NaK (45Na-55K) is 45°F; the differences in specific heat and electrical conductivity of the two eutectoids, (45Na-55K) and (56Na-44K), are small and not significant.

Lithium and sodium are not recommended in this design principally because of the severe unfreezing problem involved with each of these high-temperature-melting-point metals. The cesium-potassium eutectic is not recommended because its only advantage is the elimination of the requirement to unfreeze. Inasmuch as the unfreezing of NaK (45Na-55K) is considered a practicable procedure, NaK is chosen in order to take advantage of its higher specific heat and thermal conductivity. A strong and important general reason for choosing NaK over and above lithium and cesium-potassium is that NaK and sodium have had a variety of uses for heat transfer purposes. Advantage is taken in this feasibility study of the more refined state of the art of containing NaK as compared to the other alkali metals, except sodium.

Engine Design

Mechanical design of the basic engine follows current state-of-the-art practices. Critical areas have been checked for stress and designed to be conservative, such as bearings with a DN factor of 1.2×10^6. The critical speed of the shaft connecting the turbine and compressor was calculated to be at 41,850 rpm which is approximately 45% over the design speed of 29,000 rpm.

Compressor rotor blades are pin mounted, and the turbine rotor blades are attached by conventional fir-trees. Both of the compressor and turbine stator blades are held to the housings by circumferential slots. Construction of the compressor rotor and the intermediate shaft to the turbine rotor has been designed as an integral unit.

Attachment of the turbine rotor to the shaft is accomplished by the use of a curvic face-splined coupling. Thrust loads for this compressor-turbine unit are taken through to the front bearing package.

The free-turbine unit is supported independently through the rear housing of the exhaust diffuser and provides for the axial coupling of a reduction gear in this rear location. Lubrication of all the bearings is accomplished by a closed pressurized system.

Exhaust gases are discharged radially from the engine as the installation requires. The overall length of the engine, excluding the reduction gear, is 55 inches with a maximum frontal diameter of 27 inches. Lacking a detailed weight analysis, that the scope of this study does not include, the basic engine without regenerator is estimated to weigh 250 pounds.

The exchanger behind the compressor and the exchanger behind the turbine exit require annular-type air diffusers entering to provide the design frontal velocities of approximately 30 and 60 feet per second respectively. In designing the diffusers, emphasis has been placed on obtaining relatively uniform velocity profiles in a minimum length without incurring excessive total pressure loss. To achieve these objectives, a combination of high-rate diffusion concepts, which individually have been proven experimentally, have been incorporated in the diffuser designs. The design and analysis of diffuser between the compressor and the upstream heat exchanger has been divided into three stages. In the initial stage the design provides optimum diffusion through an area ratio of 1.77. This stage includes a curved-wall area schedule which represents a linear velocity deceleration as a function of distance at a geometric divergence angle which varies from 7½ to 10 degrees. The second stage of the diffuser has a 40-degree geometric divergence angle with an area ratio of 2.1. The outer wall of this section is made of porous steel to provide area bleed for the purpose of boundary layer removal. Approximately 3% bleed is intended at this point and will be used for bearing cooling etc. The final stage of the design is effectively a dump section with an area ratio of 2.5. Three vanes dividing four equal areas are included in this section to improve the final velocity distribution. The overall diffuser configuration has an area ratio of 9.35.

Structural considerations downstream of the turbine imposed a more restrictive set of conditions on the diffuser in that location than on the compressor exit diffuser. For this reason, there was not enough flexibility to permit incorporating several of the more attractive diffuser concepts in this design. The exit annulus from the turbine exit guide vane presents a divergent passage with a mean direction of 10 degrees outward from axial. The diffuser is required to provide an 80-degree turn to a radial heat exchanger. Three splitter vanes have been incorporated in the initial section to form 4 passages of equal divergence angles. This section of the diffuser turns through approximately 60 degrees. The remainder of the diffusion and turning is accomplished in an effective dump section.

The performance of the diffusers so described and incorporated in the design differ in detail with that assumed for purposes of computing engine performance. The performance analysis assumed 98 percent total pressure recovery in each location; this specific design implies that the total pressure recovery, behind the compressor and behind the turbine are approximately 99 and 97 percent respectively. Engine performance is a function of the product of the recoveries, approximately 96 percent in each case. Since performance is not materially affected, the original assumptions are not changed and this explanation should suffice.

1000 Horsepower Engine
Weight of Regenerator

A weight analysis of the regenerator system, including NaK, shows a total of 302.6 pounds. This total system weight, based on present state-of-the-art materials and fabrication techniques, includes the change in weight of engine components that are attributable to the addition of the regenerator system.

The weight breakdown is tabulated below:

	WEIGHT, pounds
Heat Exchanger behind compressor	
Finned tubes	72.4
Unfinned tube sections at bends	1.95
Headers and expansion chambers	17.25
Heat Exchanger behind turbine	
Finned tubes	60.8
Unfinned tube sections at bends	1.70
Headers and expansion chambers	18.67
Connecting lines	4.94
Casings, supports, shafting, and misc.	66.02
Electric heaters in headers and lines	10.00
NaK pump	30.00
NaK	18.9
Total	302.63

This total regenerator weight is added to an estimated basic engine weight of 250 pounds for a total engine-regenerator weight of 553 pounds. The specific fuel consumption of this engine, at its design point is .446 pounds per horsepower hour, as previously quoted. If the heat transfer design were thoroughly optimized on a digital computer, the weight of the heat exchangers might possibly be decreased by 10 pounds. The design items that are involved in such a refinement include fin-tube diameter ratio, fin thickness, tube spacing, tube shape and others.

As inferred in various locations in this report the weight of the heat exchangers could be decreased if certain design concepts and implied successful development work were presumed. There follows a list of design features in this category together with an estimated weight saving:

Design Feature		Weight Saving Pounds
1.	Instead of NaK use* of lithium	27.7
	sodium	15.8
2.	Use of 30 fins per inch instead of 24 in both heat exchangers	8
3.	Use of aluminum fins in heat exchanger behind the compressor	12
4.	Use of thin tapered fins** (stainless-steel-clad copper)	55
5.	Elimination of preheating system	10

All of the numbered weight saving items can be realized collectively except for the combination of Items 1 and 5.

*Including use of a 20 pound pump.

**Estimated to be available within 2 years. Before the end of 1961 fin material is expected to be developed which will save 35 pounds in the assembly.

HEAT TRANSFER CONSIDERATIONS

A heat transfer study to show reason for the choice of several values of certain parameters for the 1000-horsepower engine regenerator design was made and is reported in Appendix A. Core weight, face area and number tube layers were studied over ranges of regenerator effectiveness, percent total pressure loss and number of fins per inch. The choice of 2% pressure loss in each core is justified as representing a good compromise between weight of the heat exchanger and acceptable engine performance. The study shows that the weight of the heat exchanger core decreases continuously as a function of fin spacing up to at least 30 fins per inch. The choice of 24 fins per inch, to avoid fouling particularly behind the turbine, is justified in as much as the weight penalty is not severe down to at least 20 fins per inch. The advisability of designing down toward .50 overall regenerator effectiveness is confirmed by the rapid increase of total core weight as a function of overall regenerator effectiveness. The .004-inch copper thickness used in the design is shown to be only slightly greater than the .003 optimum demonstrated in the appendix study.

The heat transfer design calculations were made by assuming a pressure drop and the number of tube layers which fixed the face area and then by solving for the exchanger individual effectiveness. Various combinations of individual effectivenesses which give the required overall effectiveness were then weighed to find the minimum weight regenerator which meets the physical design requirements. Physical design requirements for the exchanger downstream of the compressor require that there be two tube layers per pass to keep the NaK pressure drop low and an even number of passes so that the lines connecting the exchangers are kept at the shortest possible length, except as modified by the location of the NaK pump. This combination restricts the number of tube layers in the exchanger behind the compressor to multiples of four.

In the exchanger behind the turbine the greater number of tubes per layer and the lower numbers of layers give lower NaK velocity and pressure drop allowing in turn the use of only one layer per pass. The NaK pump is located inside and toward the rear of the diffuser behind the turbine. This location leads to a requirement that there be an odd number of passes in this downstream heat exchanger.

A calculation of the design procedure is given below preceded by a table of definitions:

Definition of Symbols Used

A	Area, ft^2
C	Fluid heat capacity = WC_p
C_p	Specific heat at constant pressure, $\frac{BTU}{lb\text{-}^oF}$
D	Diameter, inches
D_{eq}	Equivalent diameter, ft = 4 (exchanger minimum free flow area) L/(total heat transfer area)
E_c	Regenerator effectiveness of exchanger behind compressor = $\frac{T_{ac2}-T_{ac1}}{T_{L2}-T_{ac1}}$
E_h	Regenerator effectiveness of exchanger behind turbine = $\frac{T_{L2}-T_{L1}}{T_{ah1}-T_{L1}}$
E_o	Overall regenerator effectiveness = $\frac{T_{ac2}-T_{ac1}}{T_{ah1}-T_{ac1}}$
f	Friction factor (see page 38)
G	Exchanger flow stream mass velocity, $\frac{lb.}{hour\text{-}ft^2}$ = ρv
h	Heat transfer film coeeficient, $\frac{BTU}{hour\text{-}ft^2\text{-}^oF}$
k	Thermal conductivity, $\frac{BTU}{hour\text{-}ft\text{-}^oF}$
L	Length of exchanger core, feet
n	Number of tubes
NTU	Number of heat transfer units = $\frac{AU}{C_{min}}$
N_{ST}	Stanton Number = $\frac{h}{GC_p}$
P	Total pressure, lb/ft^2
R	Gas constant, $ft/^oF$
Re	Reynolds number = $\frac{D_{eq}\ G}{\mu}$
t	Thickness, ft
U	Overall heat transfer coefficient, $\frac{BTU}{hour\text{-}ft^2\text{-}^oF}$
V	Velocity, ft/sec

W	Weight flow, lb/sec
y	Half the copper thickness of the fin, inches
Z	Height, inches
μ	Viscosity, $\frac{lb}{sec-ft}$
K (Ø)	Fin efficiency function, $\frac{Hour-{}^{o}F-ft^{2}}{BTU}$, (see pages 33 and 38)
ρ	density, lb/ft^3

Subscripts

a	air or gas
c	heat exchanger behind the compressor
h	heat exchanger behind the turbine
i	inside, referred to tube
L	liquid NaK
m	mean, (between inside and outside as applied to tube surface areas)
max	maximum
min	mimimum
o	overall regenerator, outside
1	entering (as applied to the liquid NaK relative to heat exchangers, "entering" refers to the heat exchanger behind the turbine).
2	leaving (as applied to the liquid NaK relative to the heat exchangers, "leaving" refers to the heat exchanger behind the turbine)

HEAT EXCHANGER DESIGN SAMPLE CALCULATIONS

A. Heat Exchanger Behind Compressor

(1) Given $P_{a1} = 14670$ lb/ft^2, $T_{am} = 1097\ ^oR$, $W_a = 8.25$ lb/sec, $W_L = 8.87$ lb/sec

$R_a = 53.345$ ft/^{o}F, $g = 32.174$ ft/sec^2, $D_{eq} = .00514$ ft,

$\frac{L}{n} = .03308$ ft., $C_{pa} = .2515$ Btu/lb-^{o}F, $\Delta P/p = .018$

$\mu_{am} = 2.085 \times 10^{-5}$ lb/sec-ft, Experimental Friction Factor Constant = .17926

$A_{flow}/A_{face} = .543$, Adjusted Stanton Correlation Constant = .0156

$Z_{fin} = .130875$ in., D_o fin = .418 in., $k_{fin} = 200$ BTU/hr-ft-^{o}F

$y_{fin} = .002$ in., $h_L = 15660$ BTU/hr-ft^2-^{o}F

$k_{tube\ wall} = 12.2$ BTU/hr-ft-^{o}F, $A_o/A_i = 15.92$, $t_{tube} = .00125$ ft.

$A_o/A_m = 14.30$, $h_{fouling(air)} = 500$ BTU/hr-ft^2-^{o}F, $A_o/_{nA\ face} = 13.971$

(2) $$\rho_{am} = \frac{P_1}{R_a T_1} = \frac{14670}{(53.345)(1097)} = .2507 \text{ lb/ft}^3$$

(3) $$nV_{max}^{1.795} = \frac{(D_{eq}^{1.205})(P_1)(\Delta P/P)(g)}{(2\frac{L}{n})(\mu_a^{.205})\text{ (Friction Factor Constant) }(\rho_a^{.795})}$$ (see page 38)

$$nv_{max}^{1.795} = \frac{(.00514)^{1.205}(14670)(32.174)(.018)}{(2)(.03308)(.0000208)^{.205}(.17926)(.2507)^{.795}} = 35640$$

(4) Select n number of tube layers = 24

(5) $$V_{max}^{1.795} = nV_{max}^{1.795}/n = \frac{35640}{24} = 1485$$

(6) $$V_{max} = \sqrt[1.795]{V_{max}^{1.795}} = \sqrt[1.795]{1485} = 58.51 \text{ ft/sec}$$

(7) $$A_{min} = \frac{W_a}{\rho_{am} V_{max}} = \frac{8.25}{(.2507)(58.51)} = .5624 \text{ ft}^2$$

(8) $$A_{face} = \frac{A_{min}}{A_{flow}/A_{face}} = \frac{.5624}{.543} = 1.035 \text{ ft}^2$$

(9) $$Re = \frac{D_{eq} V_{a_{max}} \rho_{am}}{\mu_{am}} = \frac{(.00514)(58.5)(.2507)}{2.08 \times 10^{-5}}$$

(10) $\left(\frac{1000}{R_e}\right)^{.435} = \left(\frac{1000}{3617}\right)^{.435} = .5714$

(11) N_{ST} = (Adjusted Stanton Correlation Constant) $\left(\frac{1000}{Re}\right)^{.435} = (.0156)(.5714)$

N_{ST} = .00891 (consistent with Stanton correlation J factor, see p. 38)

(12) $G_{amax} = \rho_{a_m} V_{amax} = (.2507)(58.51) = 14.67\ lb/ft^2\text{-}sec$

(13) $h_a = (3600\ C_{pa})(G_{amax})(N_{ST}) = (3600)(.2515)(14.67)(.00891)$

$h_a = 118.3\ BTU/hr\ \text{-}ft^2\text{-}{}^oF$

(14) $$K(\emptyset) = \frac{Z^2_{fin}}{(12)(k_{fin})(y_{fin})} \sqrt{\frac{D_{ofin}}{D_{ofin} - 2(Z_{fin})}}$$

$$K(\emptyset) = \frac{(.130875)^2}{(12)(200)(.002)} \sqrt{\frac{.418}{.418 - 2(.130875)}} = .00583 \text{ (see page 38)}$$

(15) $$\frac{1}{U} \quad \frac{1}{h_a} + \frac{(t_{tube})(A_o/A_m)}{k_{tube}} + \frac{A_o/A_i}{h_L} + \frac{1}{h_{\substack{fouling\\(air)}}} + \frac{K(\emptyset)}{3}$$

$$\frac{1}{U} = \frac{1}{(118.3)} + \frac{(.00125)(14.30)}{12.2} + \frac{15.92}{15660} + \frac{1}{500} + \frac{.00583}{3} = .01486 \quad \frac{hr\text{-}ft^2\text{-}{}^oF}{BTU}$$

(16) $nA_{face} = (24)(1.035) = 24.84\ ft^2$

(17) $$UA_o = \frac{(nA_{face})(A_o/nA_{face})}{\frac{1}{U}} = \frac{(24.84)(13.971)}{.01486} = 23350\ BTU/hr\text{-}{}^oF$$

(18) $C_a = (3600)(C_{pa})(W_a) = 3600\ (.2515)(8.25) = 7470\ BTU/hr\text{-}{}^oF$

(19) $C_L = 3600\ C_{pL}\ W_L = 3600\ (.234)\ 8.87 = 7470\ \frac{BTU}{hr\text{-}{}^oF}$

(20) $NTU = \frac{UA_o}{C_{min}} = \frac{23350}{7470} = 3.126$

(21) $C_L = C_a$ or $C_{min}/C_{max} = 1$

(22) $E_c = \frac{NTU}{1+NTU} = \frac{3.126}{1+3.126} = .758$ (see Reference 1, page 11, Equation 12b)

B. Exchanger Behind the Turbine

The calculation procedure for the exchanger behind the turbine is identical to that behind the compressor except that $C_{ah} > C_L$ and

$$E_h = \frac{1-e^{-NTU(1-C_{min}/C_{max})}}{1-\frac{C_{min}}{C_{max}}\ e^{-NTU(1-C_{min}/C_{max})}}$$; (see Reference 1, page 11, equation 12);

as calculated $E_h = .631$,

where $C_{min} = C_L = 7470\ \frac{BTU}{hr\text{-}^oF}$

and $C_{max} = C_{ah} = 3600\ C_{pah}\ W_a = 3600\ (.2595)\ 8.25 = 7707\ \frac{BTU}{hr\text{-}^oF}$

$\frac{C_{min}}{C_{max}} = \frac{7470}{7707} = 9642$, $C_{pah} = .2595\ \frac{BTU}{lb\text{-}^oF}$

C. Overall Regenerator

(1) $E_o = \frac{1}{\frac{1}{E_c} + \frac{1}{E_h} - 1} = \frac{1}{\frac{1}{.758} + \frac{1}{.631} - 1} = .525$

for

$C_{ah} > C_{ac} = C_L$ (see Reference 1, page 17, equations 19d or 19f)

(2) Repeat calculations for both exchangers selecting different values of n until desired E_o is obtained and the total exchanger weight is a minimum. This procedure is iterative.

D. Temperatures

given: $T_{ac1} = 959\ ^oF$, $T_{ah1} = 1479^oR$

(1) $T_{ac2} = E_o\ (T_{ah1} - T_{ac1}) + T_{ac1} = (.525)(1479-959) + 959 = 1232\ ^oR$

(2) $W_L\ C_{pL}\ (T_{L2} - T_{L1}) = W_a\ C_{pac}\ (T_{ac2} - T_{ac1})$

$T_{L2} - T_{L1} = \frac{8.25(.2515)(1232-959)}{8.87\ (.234)} = 273\ ^oR$

(3) $T_{L2} = \frac{T_{ac2} - T_{ac1}}{E_c} + T_{ac1} = \frac{1232 - 959}{.758} + 959 = 1319\ ^oR$

(4) $T_{L1} = 1319 - 273 = 1046^oR$

(5) $(WC_p \Delta T)_{ac} = (WC_p \Delta T)_L = (WC_p \Delta T)_{ah}$

$$W_a C_{pah} (T_{ah1} - T_{ah2}) = 8.25\ (.2515)\ 273 = 566.4\ \frac{BTU}{sec}$$

$$T_{ah2} = T_{ah1} - \frac{566.4}{8.25(.2595)} = 1479 - 265 = 1214\ ^{o}R$$

The results of the calculation and those items on which the heat transfer design depends are tabulated below. In addition, values for the heat exchangers designed for the 300-ST test engine are included for direct comparison later in this report.

LIQUID METAL REGENERATOR FEASIBILITY STUDY
HEAT EXCHANGER DESIGN COMPARISON

for

1000 HORSEPOWER PROTOTYPE AND 300-ST ENGINES

	1000-Horsepower Engine		300-ST Engine	
	Behind Compressor	Behind Turbine	Behind Compressor	Behind Turbine
Fin Core Material	oxygen free high conductivity copper		oxygen free high conductivity copper	
Fin Cladding Material	AMS 5521		AISI304	
Copper Thickness, inches	.004		.0034	
Cladding Thickness, inches	.002		.0033	
Tube Material	Hastelloy B		AISI304	
Tube Outside Diameter, inches	.15625		.1875	
Tube Wall Thickness, inches	.015		.020	
Fin Outside Diameter, inches	.418		.375	
Number of Fins per Inch	24		24	
Clearance, inches	.040		.050	
Pitch Between Layers-inches	.397		.368	
Staggered Pitch, inches	.452		.425	
Number of Tube Layers	24	5	16	8
Tubes per layer	72	124	10 and 11	17 and 18
Number of Passes	12	5	8	4
$A_{free\ flow}/A_{face}$	.5433		.4586	
Face Area, A_{face}, square feet	1.035	4.177	.3271	.801
Surface Area, A_o, square feet	347	291.8	54.43	66.64

	1000-Horsepower Engine		300-ST Engine	
	Behind Compressor	Behind Turbine	Behind Compressor	Behind Turbine
Surface Area Total, square feet	638.8		121.07	
Heat Transfer Coefficient Fouling Factor, $h_{fouling}$, $\frac{BTU}{hr\text{-}ft^2\text{-}{}^oF}$	500	500	500	500
Core Weight, pounds	72.4	60.8	13.1	16.4
Core Weight Total, pounds	133.2		29.5	
NaK ΔP in Exchanger, psi	17.2	20.1	17.1	8.1
NaK ΔP Total, psi	40.2		46.0	
NaK required Pumping Power, hp	15		3	
NaK Flow, gal/min	83.1		13.4	
Airflow, lb/sec	8.25		1.35	
Core Total Pressure Drop, percent	2	2	2.3	3
Entering Total Pressure, lb/ft^2	14670	2176	6139	2041
Temperature of air and Gas In, ^oR	959	1479	762	1560
Temperature Air and Gas Out, ^oR	1232	1214	1215	1128
Temperature NaK out, ^oR	1046	1319	934	1387
Reynolds Number, Re	3617	782	2564	840
Film Heat Transfer Coefficient, h_a, $\frac{BTU}{hr\text{-}ft^2\text{-}{}^oF}$	118.3	59.0	83.1	57.8
Overall Heat Transfer, Coefficient, U, $\frac{BTU}{hr\text{-}ft^2\text{-}{}^oF}$	67.3	42.7	58.4	44.6
NTU	3.126	1.668	2.642	2.471
Component Regenerator Effectiveness	.758	.631	.725	.724
Overall Regenerator Effectiveness	.525		.568	
WC_p of Air and Gas, C_a	7470	7707	1203	1261
WC_p of NaK, C_L	7470		1203	

The Stanton correlation J-factor and the friction factor, f, used in this design have been adjusted 10% downward from values plotted against Reynold Number in a Kayes & London plot (Reference 1, page 113, Figure 92). This choice of values conforms with this company's experimental data and with that in Reference 2, page 34, Figure 17.

Equation 3 is a combination and rearrangement of

$$\frac{\Delta P}{P} = \frac{4 f L \rho_a V^2}{D_{eq}(2g) P} \quad \text{and } f = (\text{friction factor constant})\,(R_e)^{-.205}$$

The former is recognized as the usual frictional pressure drop formula while the latter is empirical and consistent with friction factor data indicated above.

The fin efficiency function is adapted from Gardner's hyperbolic tangent formula (Reference 3, page 268, equation 10-8a) and is applied as a thermal resistance. It is accurate to within 1% for values of fin efficiency greater than 75%. This formula is specific for circumferential fins of rectangular cross section. The factor y as used in this design calculation is one-half the copper thickness of the fin.

The fouling factor applied to the fin surface was selected as quoted in Reference 3, page 189, Table 8-3. No estimate can be made of the time the exchanger may be used before this factor is exceeded.

Using NTU values of 3.126 and 1.668 in the exchangers behind the compressor and behind the turbine respectively under conditions of $C_{min}/C_{max} \approx 1$, counter-flow is approximated in the design. Reference 1, page 34, Figure 7 indicates a good approach to counterflow for 4 crossflow passes with unmixed flow within passes and mixed flow between passes. This design has 12 and 5 passes respectively in the exchangers behind the compressor and behind the turbine. The small clearance, triangular pitch and large number of fins per inch produce a stratification of gas flow which prevents mixing between passes as well as within passes. The serpentine arrangement of the tubes allows no mixing of the NaK. Since the effect of mixing is to reduce the effectiveness below the counterflow value, this unmixed arrangement furthers the approach to counterflow performance.

The qualitative relationships among the variables which affect size, after the matrix is selected, are as follows. Individual effectiveness increases with increasing number of tube layers and increasing face area. The face area increases with the number of tube layers since lower velocity is required at high numbers of tube layers to keep pressure drop constant. The core weight increases with both face area and number of tube layers. Thus the weight increase becomes large when high effectiveness is required. Quantitative description of these functions are found in Appendix A.

It can be seen, herein from

$$E_o = \frac{1}{\frac{1}{E_c} + \frac{1}{E_h} - 1}$$

that for constant E_o, E_c and E_h are in an inverse relationship. Since weight increases with increasing E for both exchangers there is a balance to be made between the two component regenerator effectivenesses. This balance comes about by plotting total core weight versus the ratio E_c/E_h for constant E_o. Figure 14 shows component effectiveness ratio plotted against core weight for .50, .525 and .55, overall effectiveness and the .525 design point. With E_c/E_h fixed, E_c and E_h are both fixed by the required E_o. Not every point on the curves can be met in actuality since many of them are for fractional tube layers. The absolute optimum ratio of E_c/E_h cannot be obtained. Only a small weight penalty, however, is incurred in accepting the effectiveness ratio as required by the physical limitations. The design includes an overall effectiveness of .525 which represents the closest approach that could be made to the .50 intended, within the physical limitations of the number of passes and tube layers already described.

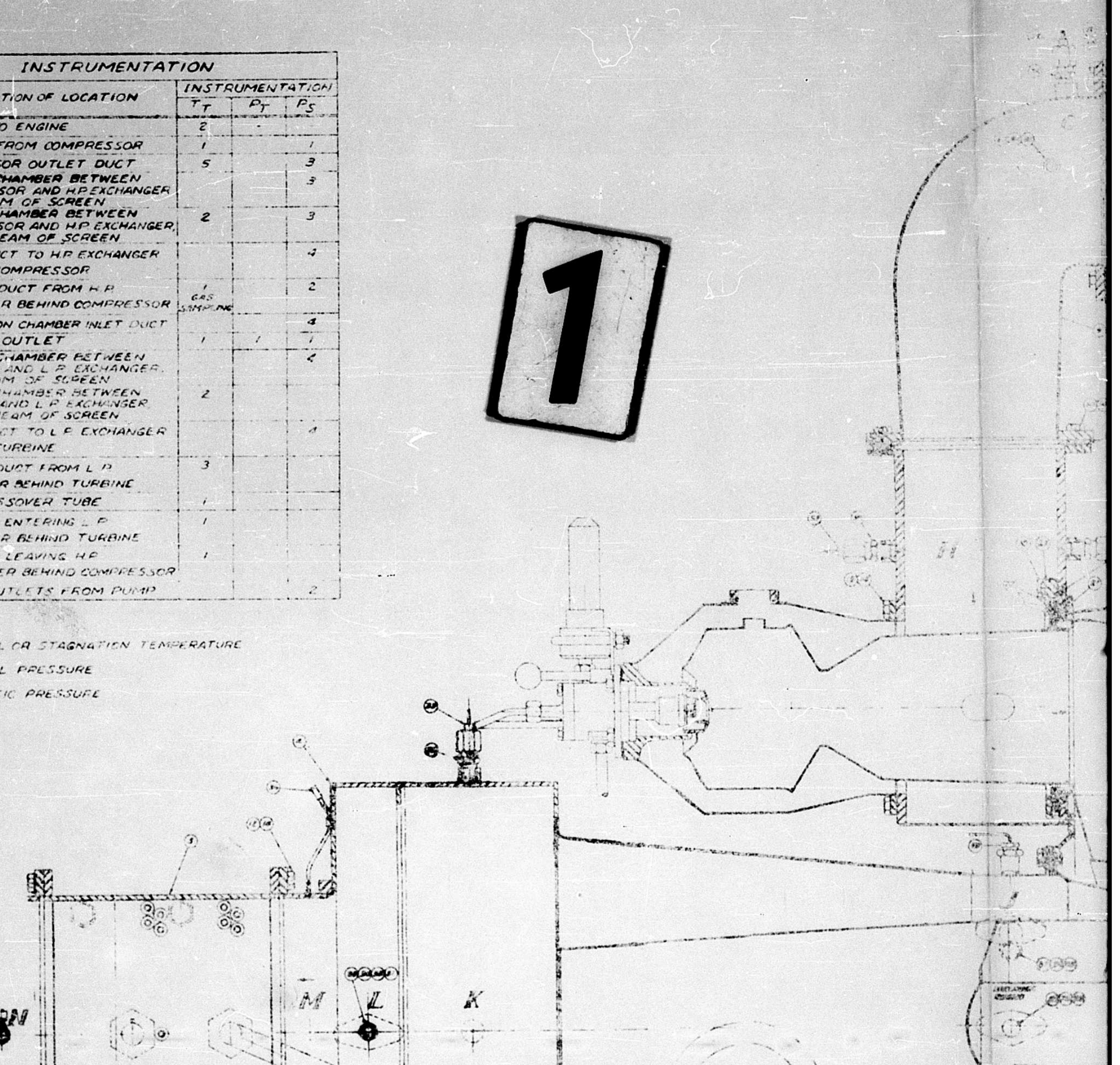
Figure 15 shows the variation in core weight as number of tube layers and face area are varied. This plot is the simplified version of the curves from which weights were determined for Figure 14. Figure 15 was constructed from the basic weight versus face area and tube layer curve for the pertinent matrix and the face area-tube layer combinations. Figure 16 shows the variation in effectiveness with face area at constant $\Delta P/P$ and the locations of the design points for each exchanger. Figure 17 shows the variation in core weight with individual exchanger effectiveness and the respective design points.

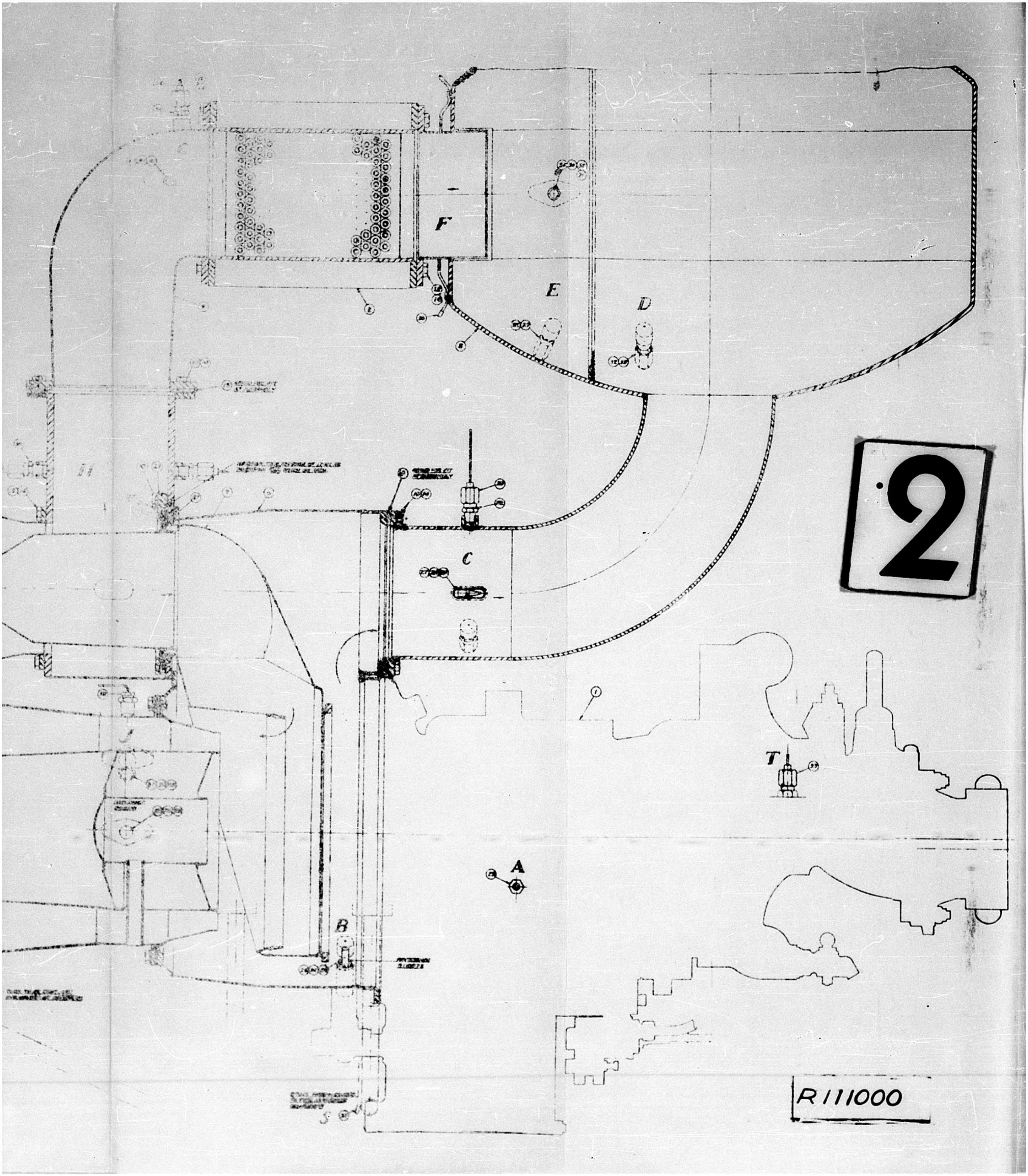
INSTRUMENTATION				
LOCATION	DESCRIPTION OF LOCATION	INSTRUMENTATION T_T	P_T	P_S
A	INLETS TO ENGINE	2	-	
B	OUTLET FROM COMPRESSOR	1		1
C	COMPRESSOR OUTLET DUCT	5		3
D	PLENUM CHAMBER BETWEEN COMPRESSOR AND H.P. EXCHANGER UPSTREAM OF SCREEN			3
E	PLENUM CHAMBER BETWEEN COMPRESSOR AND H.P. EXCHANGER, DOWNSTREAM OF SCREEN	2		3
F	INLET DUCT TO H.P. EXCHANGER BEHIND COMPRESSOR			4
G	OUTLET DUCT FROM H.P. EXCHANGER BEHIND COMPRESSOR	1 GAS SAMPLING		2
H	COMBUSTION CHAMBER INLET DUCT			4
J	TURBINE OUTLET	1	1	1
K	PLENUM CHAMBER BETWEEN TURBINE AND L.P. EXCHANGER, UPSTREAM OF SCREEN			4
L	PLENUM CHAMBER BETWEEN TURBINE AND L.P. EXCHANGER, DOWNSTREAM OF SCREEN	2		
M	INLET DUCT TO L.P. EXCHANGER BEHIND TURBINE			4
N	OUTLET DUCT FROM L.P. EXCHANGER BEHIND TURBINE	3		
P	NaK CROSSOVER TUBE	1		
R	NaK LINE ENTERING L.P. EXCHANGER BEHIND TURBINE	1		
S	NaK LINE LEAVING H.P. EXCHANGER BEHIND COMPRESSOR	1		
T	WATER OUTLETS FROM PUMP			2

T_T - TOTAL OR STAGNATION TEMPERATURE

P_T - TOTAL PRESSURE

P_S - STATIC PRESSURE

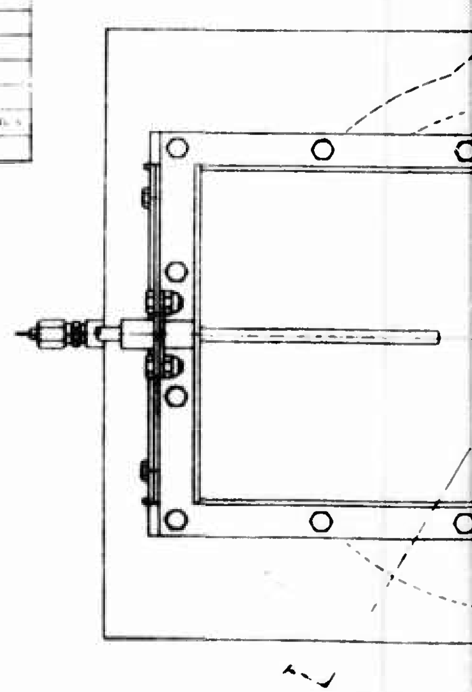



F
E
D
C
2
T
J
A
B
R 111000

1

	ITEM NO	PART NO	REFERENCE	DESCRIPTION	DWG REQ	QUAN	MATERIAL OR SPEC
	40	AN4CH7A		BOLT-MACHINE AIRCRAFT		7	
B	39	MS20500-428		NUT-SELF-LOCKING HEX 1200°F		7	
	38	R533272		PLUG-STATIC PRESSURE		1	
	37	WAD 304613-12	REWORKED PER ENGINEER	PROBE-TOTAL PRESSURE		2	
	36			SPAGHETTI		AS REQD	
	35	AN3630428		NUT-SELF LOCKING		27	
B	34	AN4C5A		BOLT-MACHINE, AIRCRAFT		25	
	33	R533271		CAP-STATIC PRESSURE		21	
	32	AN816-4C		NIPPLE-FLARED TUBE & PIPE THD		19	
	31	WAD 306692		ADAPTER AJUSTABLE MOUNTING		7	
	30			THERMOCOUPLE WIRE WITH PMSS PLUG		3	CHROMEL ALUMEL
	29	WAD 302854Z		THERMOCOUPLE-BARE WIRE L P		2	
	28	BG 16720-1X1		-GAS SAMPLING		1	
	27	WAD 313640-1		RAKE T/C 5 UNIT		1	
	26	WAD 306743-2		-STAGNATION TYPE		1	
	25	WAD 306743-3		-STAGNATION TYPE		1	
A	24	WAD 306743-8		THERMOCOUPLE-STAGNATION TYPE		7	
	23	RIP-2		TABWASHER		3	
	22	R533246		RING CLAMP		1	
	21	R533245		RING SEAL		1	
	20	AN819-4C		SLEEVE-COUPLING		8	
	19	AN818-4C		NUT-COUPLING		8	
	18	AN815-4C		UNION-FLARED TUBE		4	
A	17						
	16	R533242		CROSSOVER TUBE-EXCHANGER		1	
	15	R533241		SHIM-BURNER INLET		1	
	14	MS20995C32		LOCKWIRE		AS REQD	
A	13	R533268		DUCT-L P EXHAUST		1	
B	12	AN4CH4A		BOLT-MACH, AIRCRAFT		113	
	11		JOHNS MANVILLE STYLE NO. 2559	PACKING .187 × .187 × 36" TOTAL LG		1	
B	10	AN4CH6A		BOLT-MACH, AIRCRAFT		39	
	9	R111016		DUCT-H P EXIT		1	
	8	R533240		SHIM-COMPRESSOR EXIT		1	
	7	R111004		DUCT REWORK-BURNER TO TURBINE		1	
	6	R111003		HOUSING REWORK-COMPRESSOR		1	
	5	R111002		DUCT COMP TO HP EXCHANGER		1	
	4	R111001		DUCT L P INLET		1	
	3	R111009		HEAT EXCHANGER-L P		1	
	2	R110998		HEAT EXCHANGER-H P		1	
	1			ROVER ENGINE		1	

PARTS LIST



SCALE FULL SIZE UNLESS OTHERWISE SPECIFIED

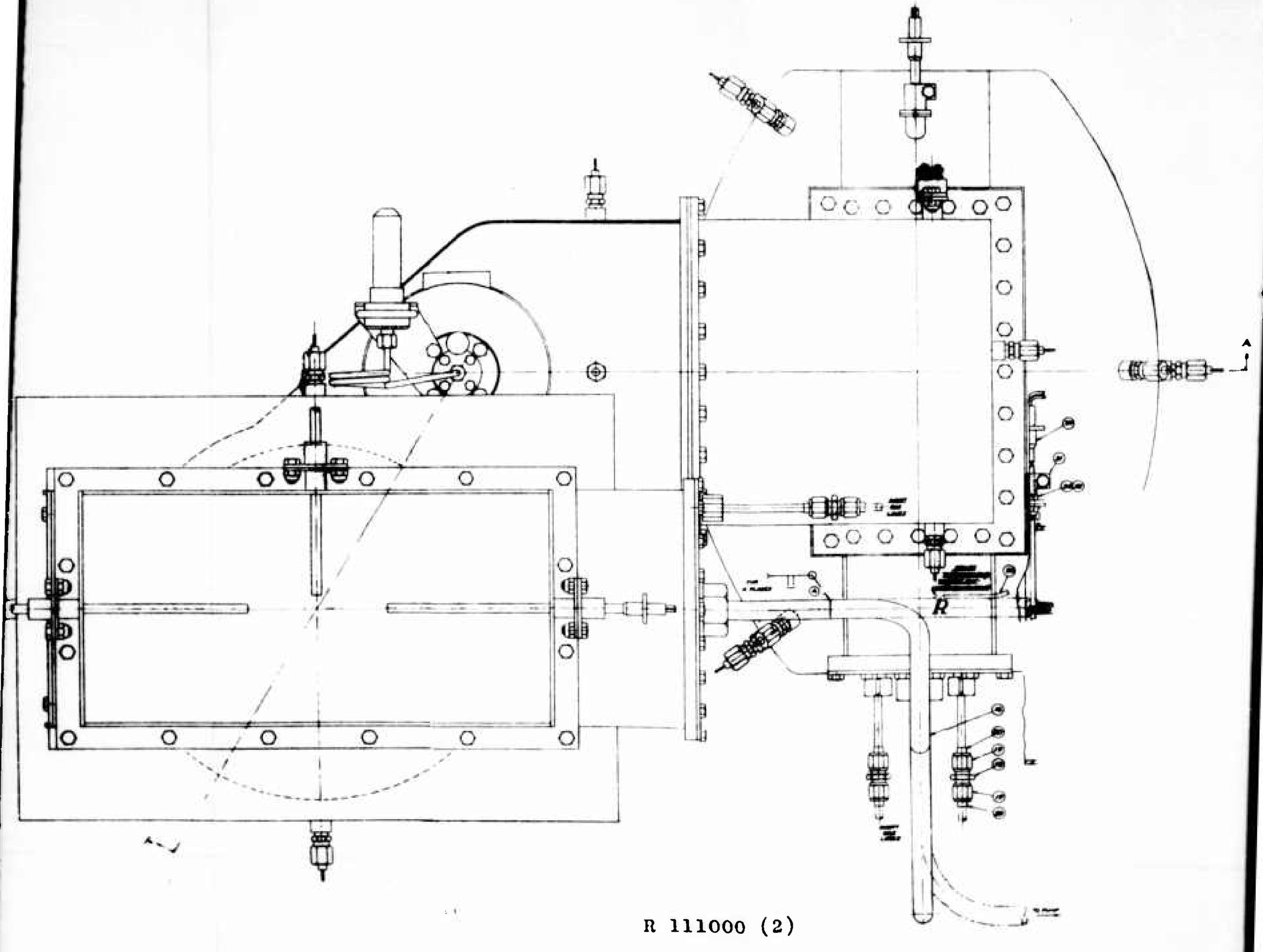
DFTSM	CKR	CHIEF DFTSM	DESIGN ENGR	MODEL ENGR	
					SPECIFICATIONS
					SEE B/M
MET	PROCESS ENGR	ENGR	ENGR	ENGR	MAT'L SPECS

CURTISS WRIGHT CORP
RESEARCH DIVISION
QUEHANNA, PA

HEAT EXCHANGER TEST UNIT

R111000

2



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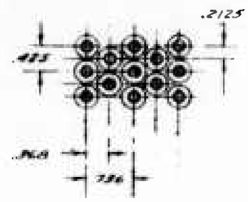
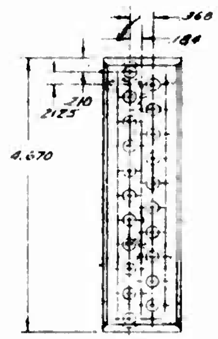


SECTION E-E



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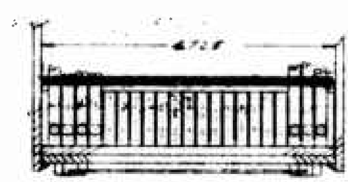

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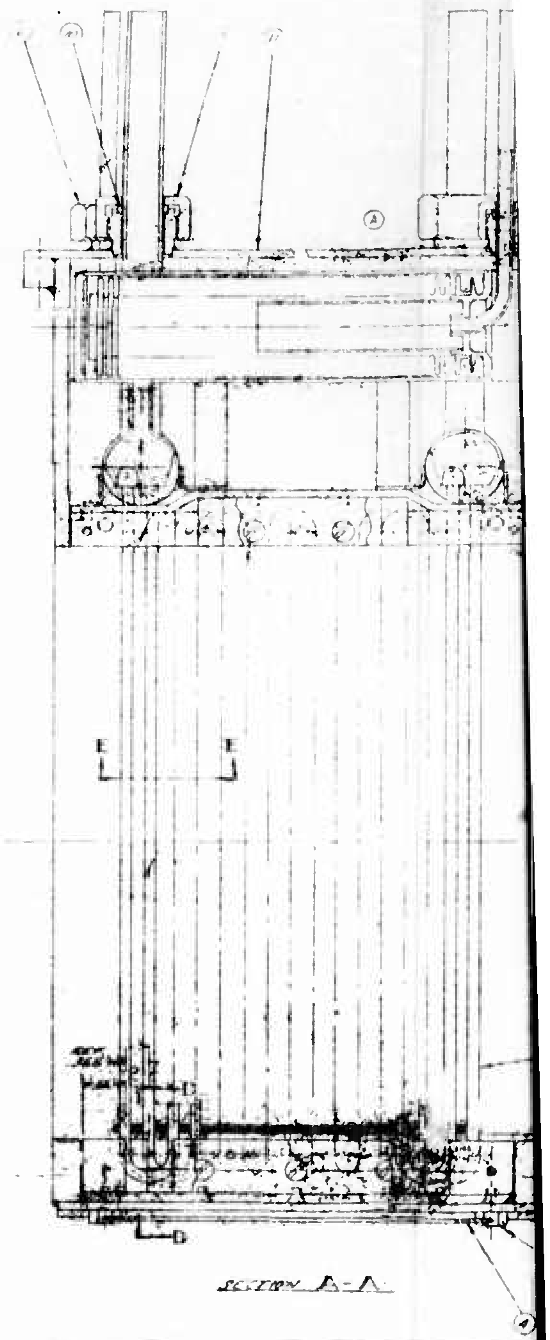

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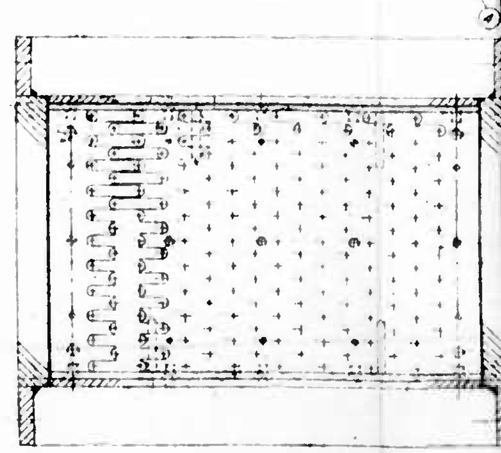

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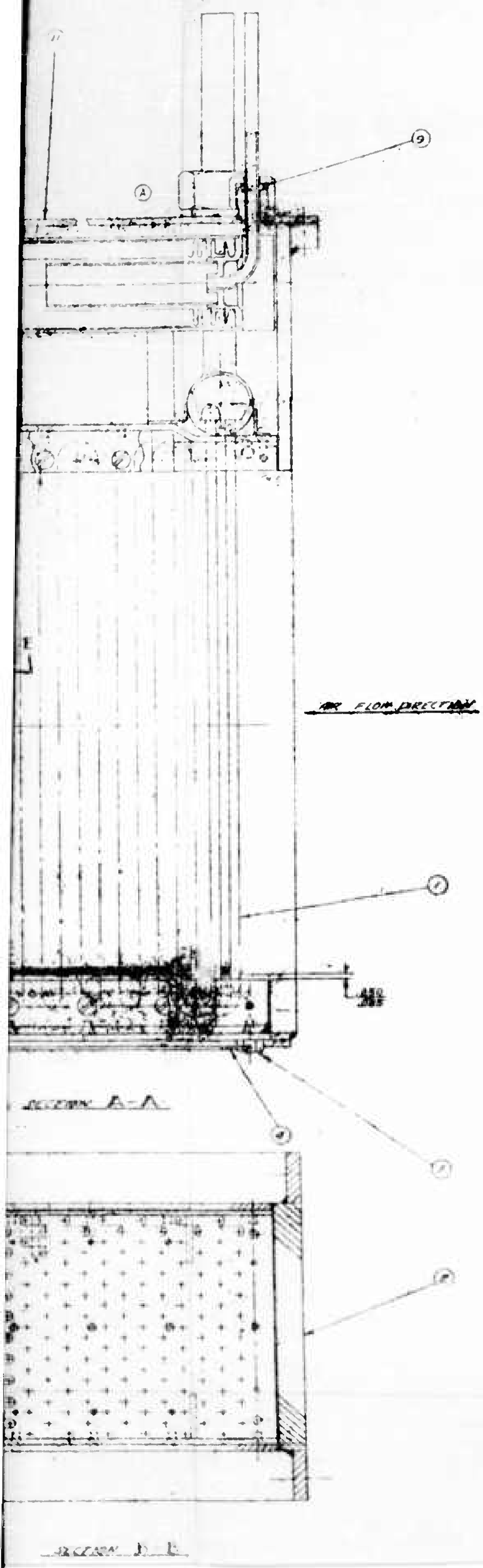


2

ITEM NO.	PART NO.	REFERENCE	DESCRIPTION	DWG. REQ.	QUAN.
11	R111023		END PLATE ASSY.- H.P. HEAT EXCHANGER		1
10	AN900-10		GASKET-COPPER ASBESTOS		2
9	AN900-4		GASKET-COPPER ASBESTOS		2
6	R533226		NUT, HEAT EXCHANGER PACKING		2
5	R533227		NUT, HEAT EXCHANGER PACKING		2
4	MS20995C32		WIRE, LOCK		AS REQ'D
3	AN106506		BOLT-CRES DRILLED HEX HD.		34
2	R111007		HOUSING ASSY.- H.P. HEAT EXCHANGER		1
1	R111010		CORE ASSEMBLY-H.P. HEAT EXCHANGER		1

PARTS LIST

DRFTM.	CKR.	CHIEF DRFTM.	DESIGN ENGR.	MODEL ENGR.
MET.	PROCESS ENGR.	ENGR.	ENGR.	ENGR.

SPECIFICATIONS

MAT'L. SPECS.

BREAK SHARP EDGES
FILLETS & RADII — APPROX.
FINISHED DIMENSIONS
ANGLES
CONCENTRICITY — F.I.R.
SURFACE FINISH — √
UNLESS OTHERWISE SPECIFIED

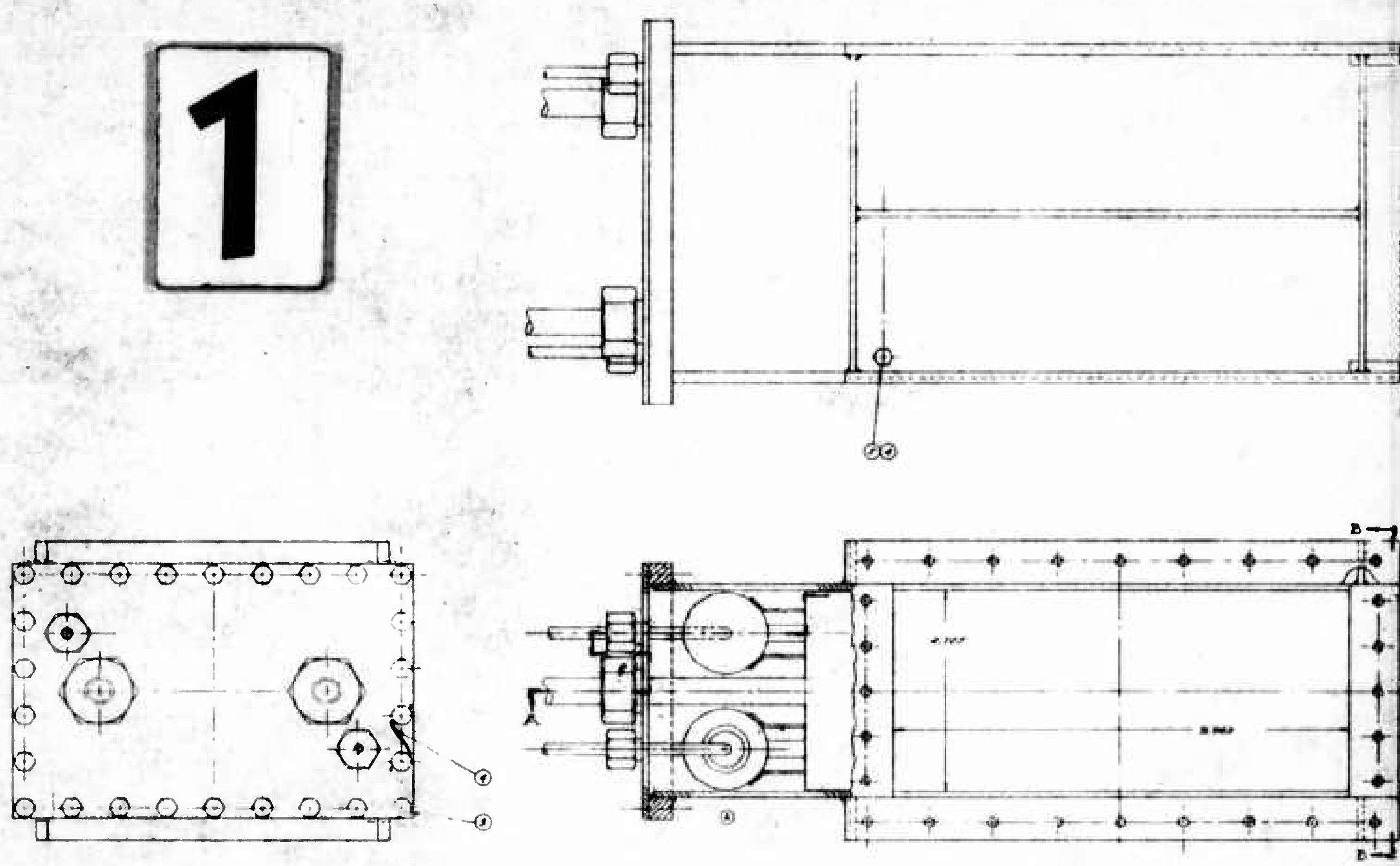
CURTISS-WRIGHT CORP.
RESEARCH DIVISION
QUEHANNA, PA.

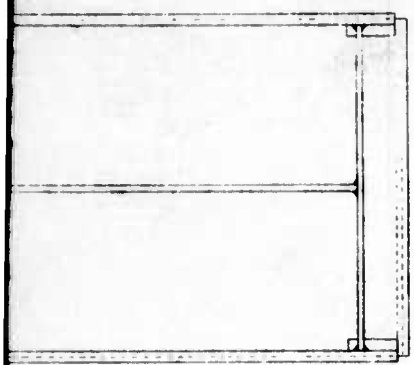
HEAT EXCHANGER-
HIGH PRESSURE

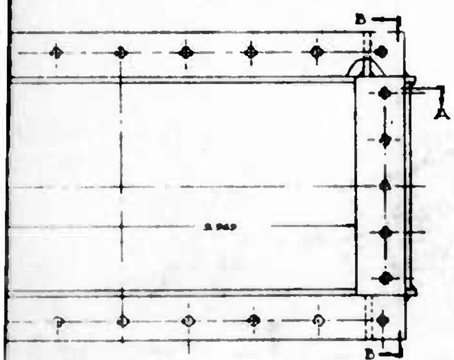
R110998

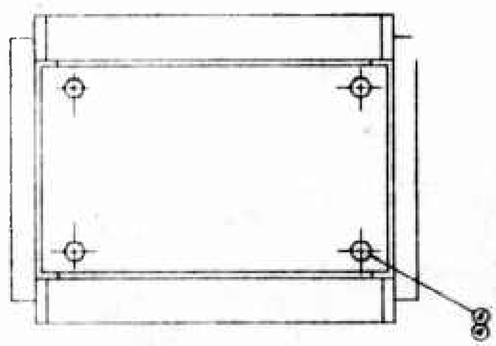
STANPAT CO. WHITESTONE N.Y. U.S.A.

1



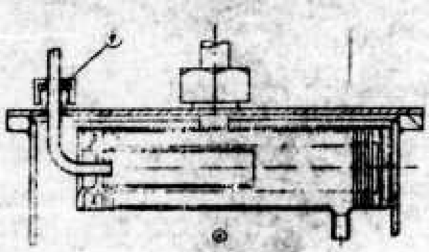






2

R 110998 (2)



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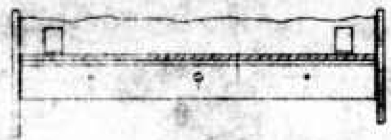

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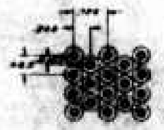

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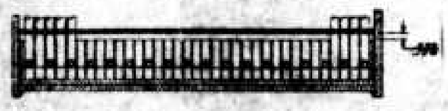
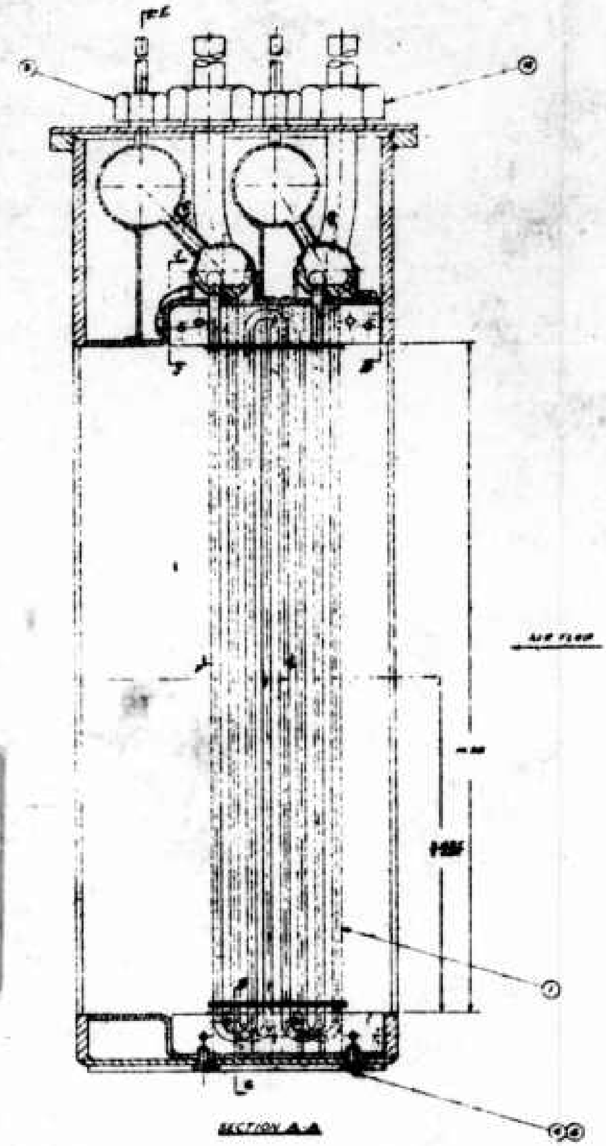


SECTION B-B

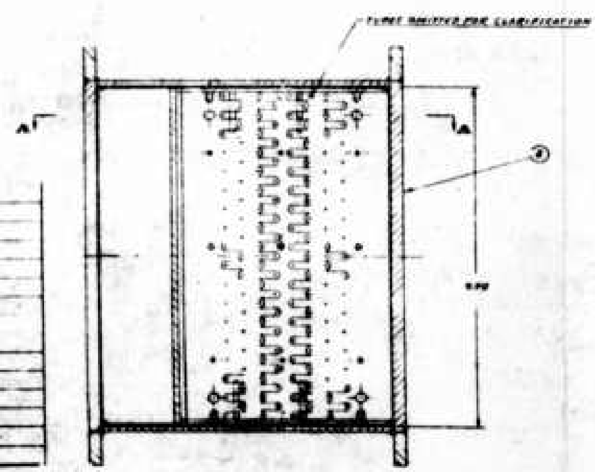


SECTION A-A

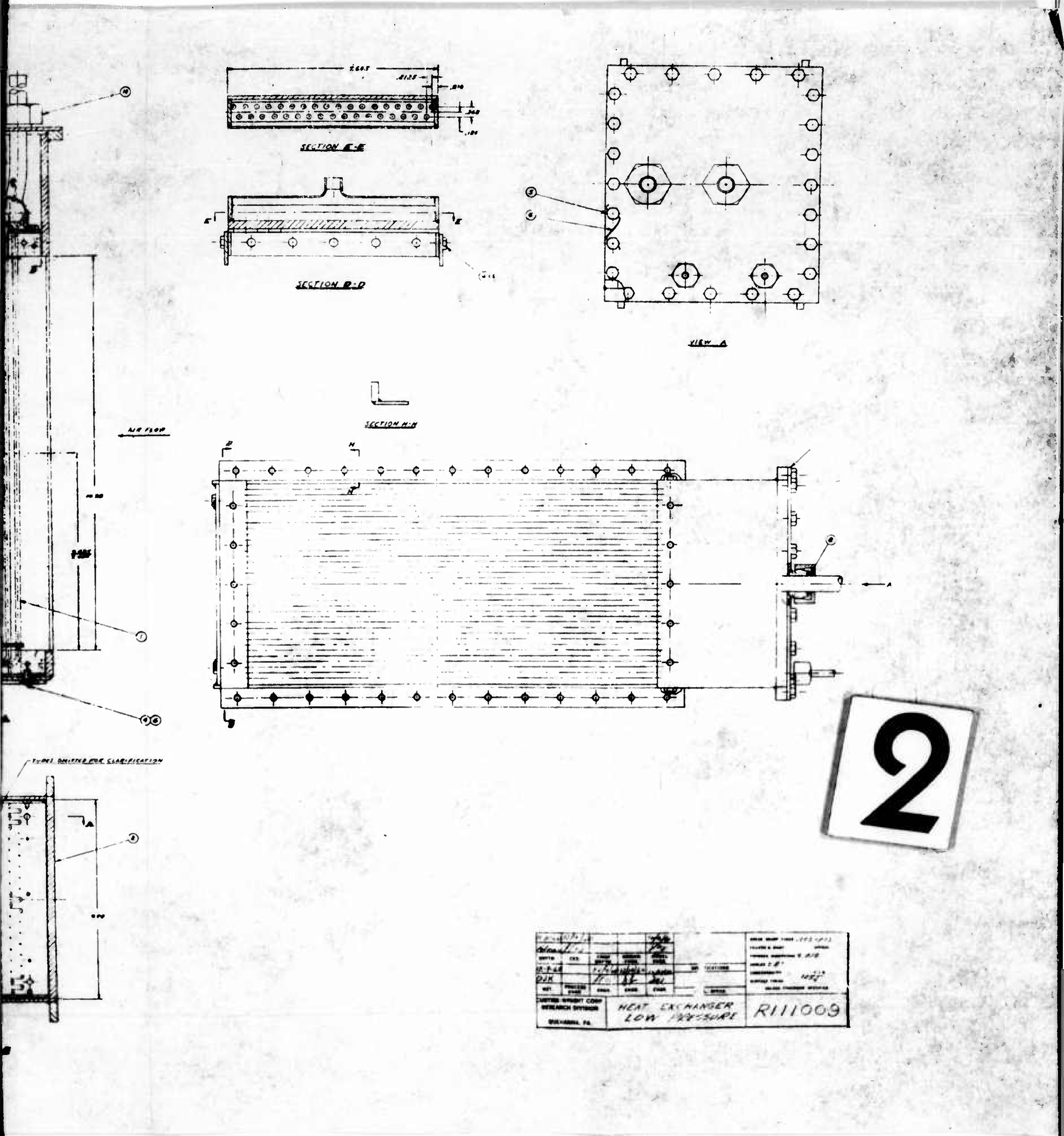
1

ITEM NO.	PART NO.	REFERENCE	DESCRIPTION		REQ.	MATERIAL OR SPEC'S
10	RS71227		NUT-HEAT EXCHANGER PACKING		2	
9	RS53024		NUT-HEAT EXCHANGER PACKING		2	
8	AN900-[illegible]		GASKET-COPPER ASBESTOS		2	
7	AN900-6		GASKET-COPPER ASBESTOS		2	
6	[illegible]		WIRE LOCK		AS REQ'D	
5	AN[illegible]		BOLT-CRES DRILLED HEX HD.		[illegible]	
4	AN[illegible]		BOLT-CRES DRILLED HEX HD.		6	
3	R111022		END PLATE ASSEMBLY [illegible]		1	
2	R111013		HOUSING ASSEMBLY [illegible]		1	
1	R111015		CORE ASSEMBLY [illegible]		1	

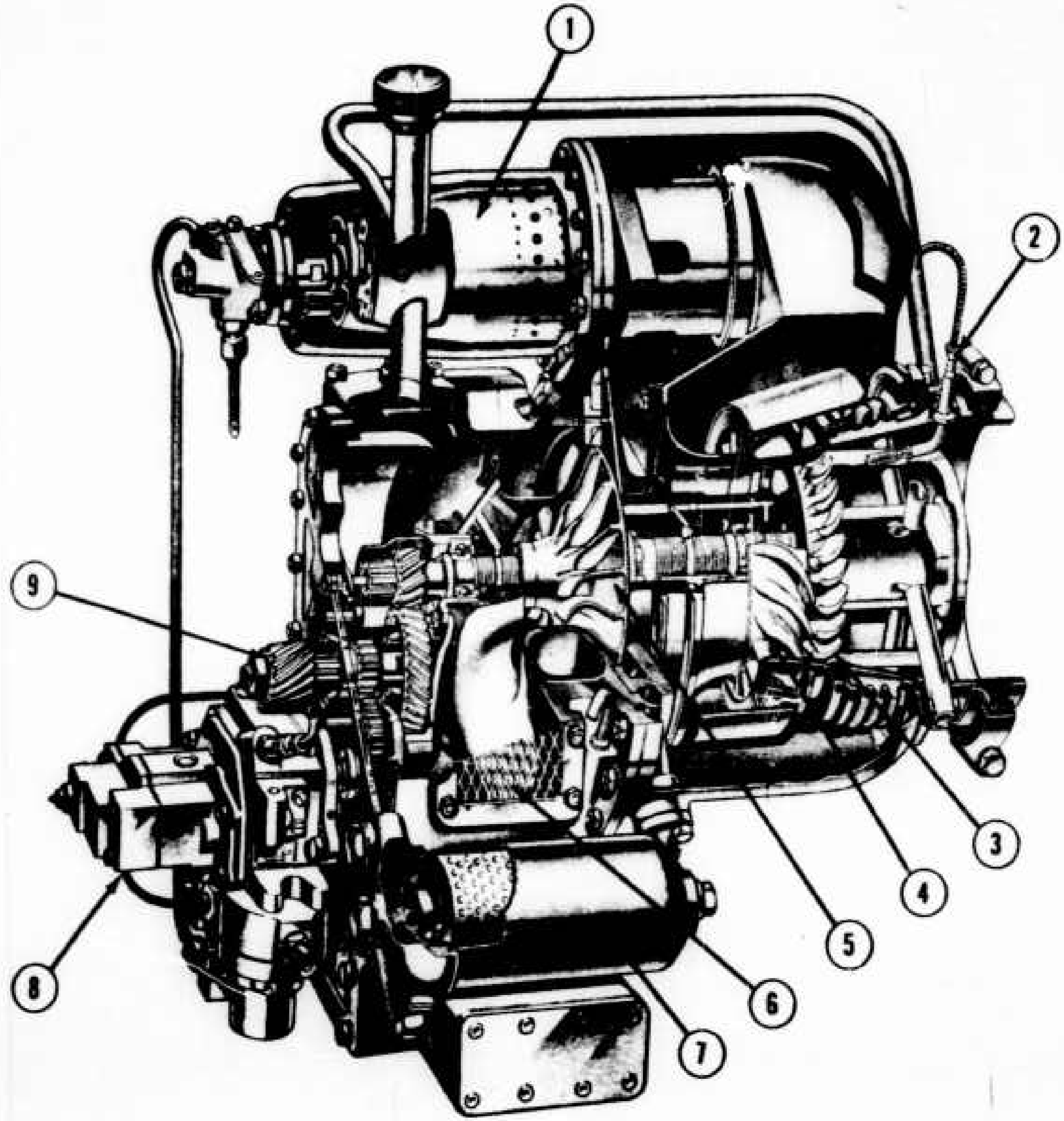
PARTS LIST



SECTION D-D

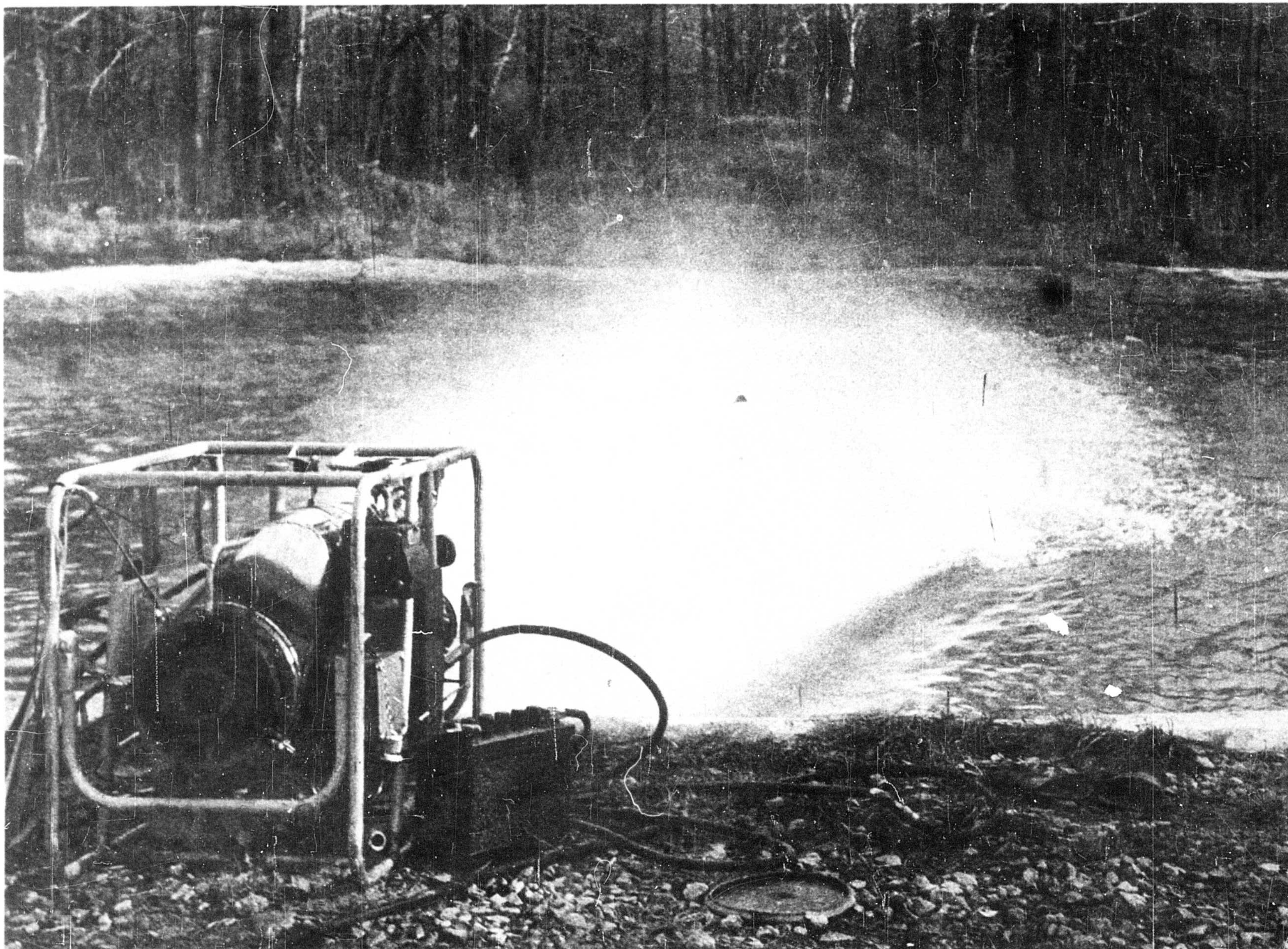

SECTION E-E
SECTION D-D
VIEW A
SECTION H-H
AIR FLOW
TUBES OMITTED FOR CLARIFICATION
2
CURTISS-WRIGHT CORP
RESEARCH DIVISION
QUEHANNA, PA.
HEAT EXCHANGER
LOW PRESSURE
R111009

300-ST TURBOSHAFT ENGINE



1- REVERSE FLOW COMBUSTION CHAMBER
2- EXHAUST TEMPERATURE SENSOR
3- TURBINE
4- TURBINE NOZZLE
5- IMPELLER
6- AIR INTAKE
7- PRESSURE OIL FILTER
8- FUEL CONTROL UNIT
9- OUTPUT PINION

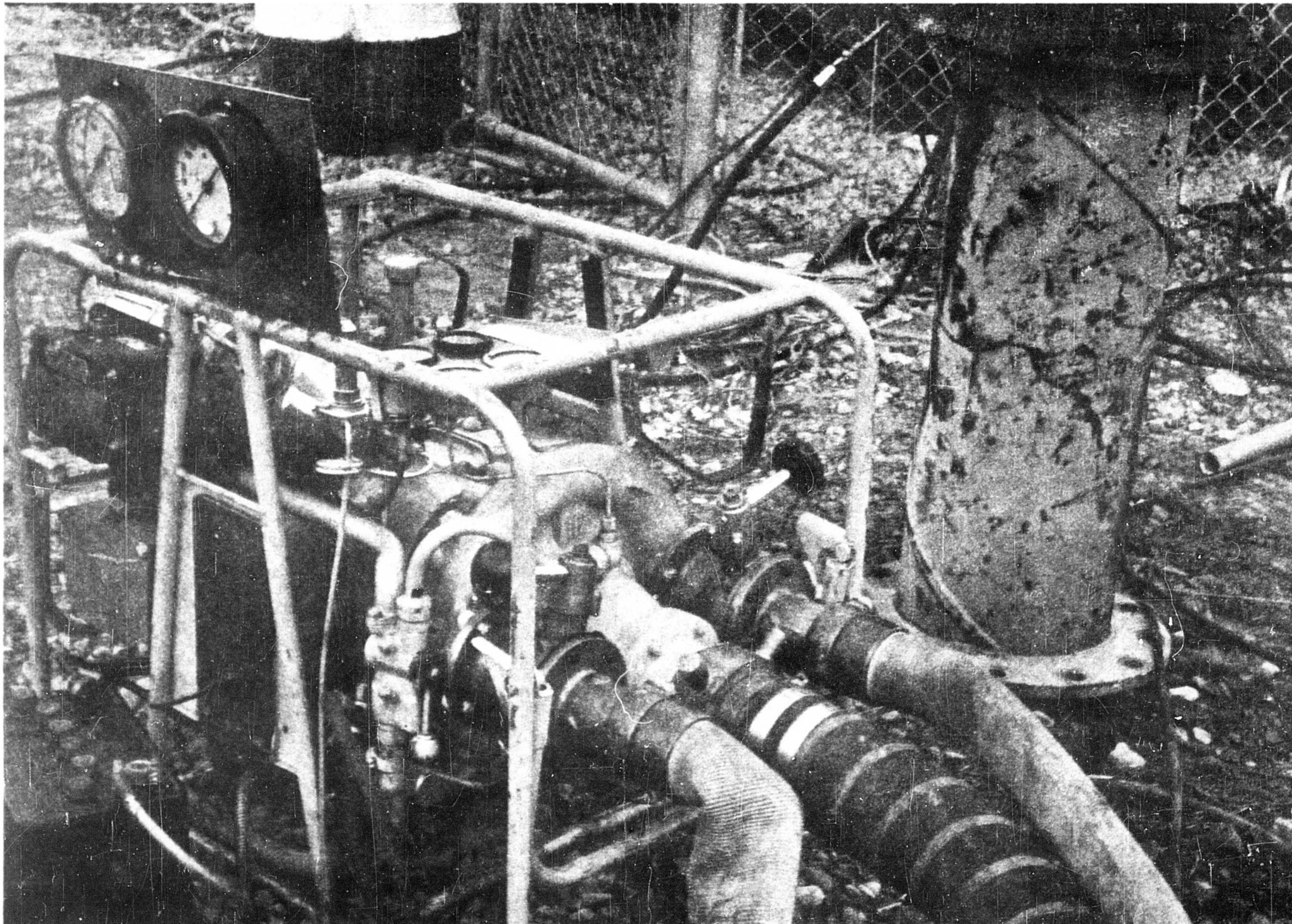
Series 300-ST Turboshaft Engine



Engine Calibration Test with Pump Discharging Directly Into Pond - Rear View

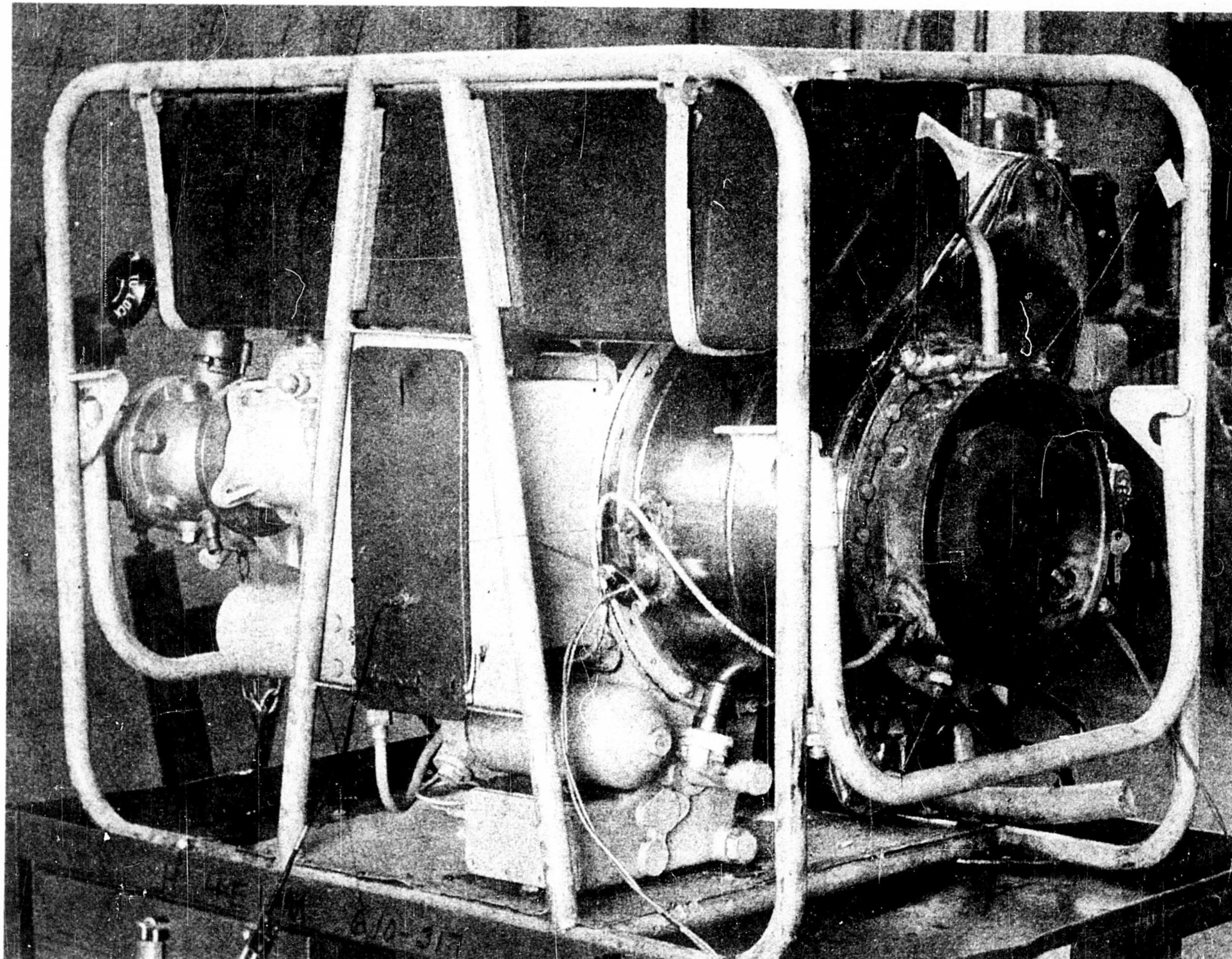
Series 300-ST Turboshaft Engine

Engine Calibration Test with Discharge Hoses Directed
Away from Pond



Engine - Pump Assembly with Test Equipment Attached.

Series 300-ST Turboshaft Engine



Engine - Pump Assembly - Side View

SERIES 300-ST TURBOSHAFT ENGINE

The 300-ST engine is a small, centrifugal-flow-compressor-type gas turbine rated at a nominal 60 horsepower. The basic engine weighs 89 pounds; its height is 24 inches; width, 15 inches; length, 19 inches. The engine is compact and features a single reversed-flow combustion chamber mounted semi-externally out of the front of the engine. A centrifugal water pump is incorporated in the engine, geared to the drive shaft.

Air enters at two side intakes, passes through a single-sided centrifugal impeller into a radial diffuser and in turn enters an annular type plenum chamber which envelopes the hot-gas passage from combustion chamber to turbine. Leaving the annular plenum chamber the air enters the combustion chamber, reverses its flow as fuel is burned and passes in the above-mentioned hot-gas passage to the turbine, from there exhausted axially. A cut-away of the engine (page 45) shows the pertinent physical features. A fuller description of the engine is found in Reference 5.

The basic engine has been calibrated at Quehanna, Pennsylvania. This calibration is reported under separate cover in TREC Technical Report 61-47, Reference 4. Photographs of the engine have been taken from Reference 4 and are reprinted herein on pages 46 thru 49. For a description of the calibration running and its results, reference should be made to TREC Technical Report 61-47.

Drawing R111000 (page 40) shows the regenerator as assembled on the Series 300-ST turboshaft engine. Drawing R110998 and R111009 (pages 42 and 44) show the exchanger assemblies behind the compressor and behind the turbine respectively. This engine is well suited for adaptation to regenerator testing. The compressor delivery air is readily routed through a heat exchanger and back into the combustion chamber with a minimum of rework to the engine. Turbine exhaust gases continue without change in direction through an axial heat exchanger placed downstream of the turbine. As shown on drawing R111000, the reworks to the engine include the following basic changes:

1. Removal of the combustion chamber from an internal and forward-facing position to an external aft position. This change involves cutting of a circular hole in the upper rear end of the compressor outlet housing and welding on of a new combustion-chamber-air-inlet-annulus-attaching flange as well as removal of the top of the turbine inlet ducting and installation of a transition piece from the rectangular turbine inlet duct to the circular burner outlet.

2. Ducting of compressor delivery air from the original combustion chamber location through the exchanger to the side of the new combustion chamber inlet housing.

3. Ducting of outlet gas to the turbine exchanger through a transition that is welded to the existing tail pipe.

The ducting of the two exchangers is designed to maintain gradual turns and low velocity for low pressure drop in the circuit. Compressor delivery air and the gas leaving the turbine are dumped at low velocity into plenum chambers and are passed through screens before entering the respective heat exchangers in order to destroy large-scale turbulence. The controlling purpose of the design is to establish uniform distribution entering the heat exchangers. Stress in the ducting is small, as allowed by the moderately low pressure. The ducting material AISI 304 selected on the basis of 1000-hour operation for creep, rupture and corrosion is more than adequate for the testing proposed for this engine.

The instrumentation for both the engine and the regenerator is shown on drawing R111000 (page 40). A list of the instruments by location follows:

Instrumentation

Position	Location	Instruments
A	Inlets to engine	2 thermocouples
B	Outlet from compressor	1 total temperature thermocouple 1 wall static pressure tap
C	Compressor outlet duct	5 total temperature thermocouples (rake) 3 wall static pressure taps
D	Plenum chamber between compressor and exchanger, upstream of screen	3 wall static pressure taps
E	Plenum chamber between compressor and exchanger, downstream of the screen	3 wall static pressure taps 2 thermocouples
F	Inlet duct to exchanger behind compressor	4 wall static pressure taps
G	Outlet duct from exchanger behind compressor	1 total temperature thermocouple probe(gas sampling) 2 wall static pressure taps
H	Inlet duct to combustion chamber	4 wall static pressure taps
J	Outlet from turbine	1 total temperature thermocouple 1 total pressure probe 1 wall static pressure tap

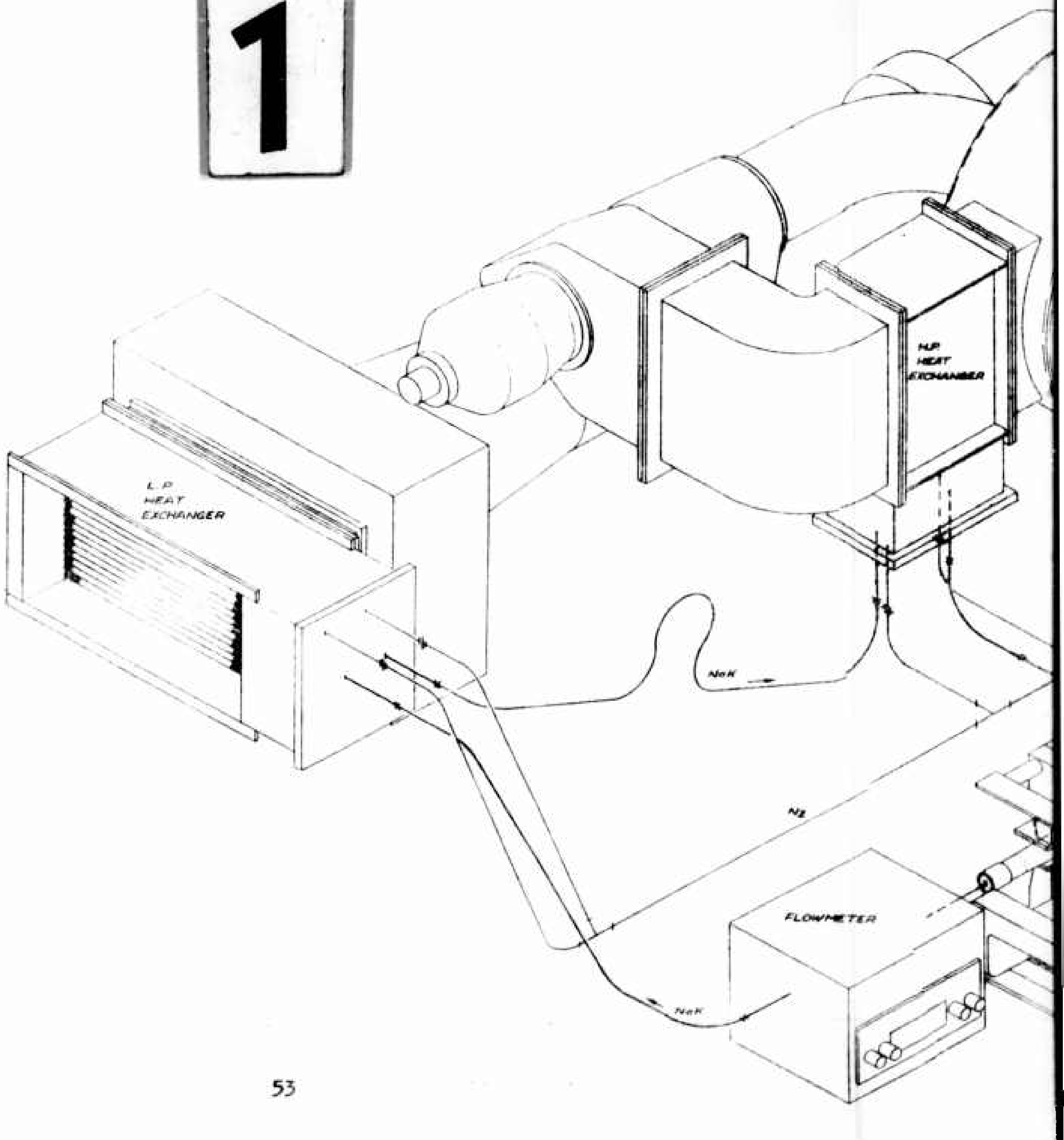
Position	Location	Instruments
K	Plenum chamber between turbine and exchanger, upstream of screen	4 wall static pressure taps
L	Plenum chamber between turbine and exchanger, downstream of screen	2 thermocouples
M	Inlet duct to exchanger behind turbine	4 wall static pressure taps
N	Outlet duct from exchanger behind turbine	3 total temperature thermocouples
P	NaK cross-over tube	1 thermocouple
R	NaK line entering exchanger behind turbine	1 thermocouple
S	NaK line leaving exchanger behind compressor	1 thermocouple
T	Water outlets from pump	2 pressure taps

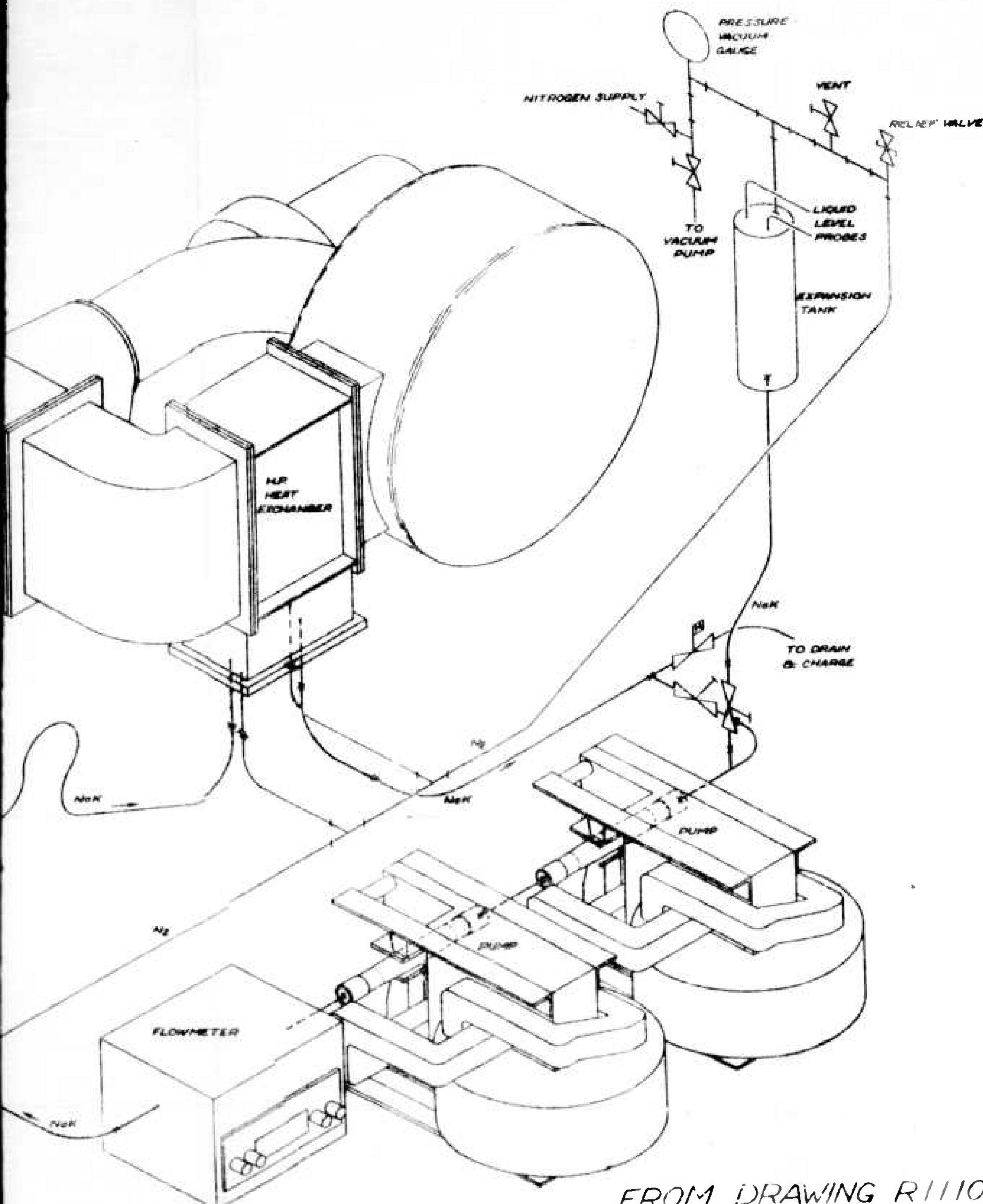
Design of 300-ST Heat Exchangers

The purpose of incorporation of a regenerator in the 300-ST engine is to demonstrate performance of a light-weight, liquid-metal regenerator system. The designs of 300-ST exchanger cores parallel the 1000-horsepower core designs and include the use of the similar finned tubes, headers and expansion chambers incorporating bellows. The calculation procedure used to design the 300-ST exchangers is the same as that for the 1000-horsepower design. Certain deviations were taken in materials and other details to expedite procurement and minimize cost.

The major dimensional differences between the two designs are the fin outside diameter, tube outside diameter and tube wall thickness. 3/16-inch tubing with a wall thickness of .020 inches, as used, is more readily available than the 5/32-inch size and conforms to the current tooling of the finned-tube fabricators. .0034-inch copper fin material clad with .0033-inch stainless steel is in turn more available at this time than the .004-inch copper with .002-inch stainless steel as specified in the prototype core. The tube material and the fin cladding have been changed to AISI 304 to accomplish considerable savings in cost, mainly by eliminating expensive inspection.

1





2

FROM DRAWING R111038

SERIES 300-ST HEAT EXCHANGERS

	Exchanger Behind Compressor	Exchanger Behind Turbine
Number of finned-tube layers in direction of airflow	16	8
Number of passes of each serpentined tube	8	4
Tubes per layer across air stream	10 and 11	17 and 18
Configuration of flow passage	Rectangular 4.705 inches x 9.97 inches	Rectangular 14.97 inches x 7.68 inches
Face Area	47.10 square inches	115.34 square inches
Finned Tube O.D.	.1875 inches	.1875 inches
Finned Tube wall thickness	.020 inches	.020 inches
Fins per inch	24	24
Fin installation configuration	Helical	Helical
Fin thickness	.01/.008 inches	.01/.008 inches
Fin configuration	.0034 inches copper .0033 inches stainless clad on both sides	.0034 inches copper .0033 inches stainless clad on both sides
Fin O.D.	.375 inches	.375 inches
Tube material	AISI 304	AISI 304
Fin material	Copper, cladding AISI 304	Copper, cladding AISI 304
Header material	AMS 5560	AMS 5560
Expansion chamber material	AMS 5560	AMS 5560
Expansion bellows material	AMS 5560	AMS 5560
Housing material	AISI 304	AISI 304

The reduced amount of thermal unbalance of the 300-ST exchanger as compared to the 1000-horsepower design was made necessary by the physical requirement of NaK pressure drop, which indicated that there be two layers per pass, and by the requirement of an even number of passes to keep the lines short and with the headers properly located. Percent core pressure drops are calculated to be 2.3% and 3% in the heat exchangers behind the compressor and behind the turbine respectively.

A tabulated comparison of the 300-ST and 1000-horsepower designs, as previously introducted, is made on page 36. In addition there follows on page 54 a tabular description of the 300-ST heat exchangers which supplements and in some cases repeats the description as tabulated in the other figure.

The effect on core weight of increasing the tube outside diameter from 5/32 to 3/16 inches and the tube thickness from .015 to .020 inches and of decreasing the fin outside diameter from .418 to .375 inches has been calculated for the 300-ST design. For identical regenerator performance these dimensional differences result in an increase of 4% in core weight.

The relaxation in quality of material for the containment of NaK is justified since no endurance testing is required. The regenerator is considered expendable after 10 to 20 hours of performance testing. The material specified are considered adequate to accomplish this performance testing.

The bellows in each expansion chamber is designed to allow for 3.75 cubic inches displacement; in the 4 expansion chambers, 15 cubic inches total displacement. The volume of the entire NaK system without expansion chambers is 123 cubic inches. When the external expansion chamber (riser) is valved shut, the internal expansion chambers have adequate capacity to handle the expansion and contraction of the NaK in the system.

<u>NaK Circulating Loop System</u>

Schematic drawing from R 111038 (page 53) shows the NaK circulating loop system. This basic system includes a pump, flowmeter, expansion tank and auxiliary equipment consisting of valves, piping and temperature and pressure gages. A laboratory vacuum pump will be attached when charging the system. This minimum NaK loop concept should adequately support at least 20 hours of heat exchanger performance testing.

After connection to the engine regenerator units, the system will be charged with NaK by first filling the entire system with nitrogen, by then evacuating the system, and finally by allowing the NaK to rise into the partial vacuum. The liquid level indicators in the expansion tank will show when the system is full. System conditions will be accomplished by charging bulk NaK, heating

and dumping. A cover of inert gas under pressure will be kept on the free NaK surface in the expansion chamber. During a portion of the 300-ST engine testing the valves in the line to the external expansion tank will be closed to demonstrate operation of the flight-type expansion chambers incorporating bellows. The drain valve will have a remote control feature to make possible the dumping of the major portion of the NaK in the system without delay at any time. Two alternating current electro magnetic pumps will be used to flow the NaK. The voltage will be controlled to the pumps by Variacs capable of supplying 30 amps at 270 volts. NaK flow will be measured by an electrodynamic flowmeter. The expansion tank is located at the high point of the loop at the suction side of the pump. It is designed to contain expansion of all liquid metal from ambient temperature to the top operating temperature. The tank design includes upper and lower liquid level indicators to monitor the liquid metal level, a cover of inert gas, mentioned before, with a pressure gage and a shut-off valve and a pressure relief valve. The NaK loop itself will be fabricated from 5/8-inch AISI 304 tubing.

REFERENCES

1. Kayes, W.M. and London, A.L., Compact Heat Exchangers, McGraw-Hill Book Company, 1958.

2. Carnavos, T.C., Heat Transfer and Pressure Loss Performance of Griscom-Russell Special Small K-Fin Helically Finned Tubes, RT-2-58, The Griscom-Russell Company, May, 1958.

3. McAdams, W.H., Heat Transmission, McGraw-Hill Book Company, 1954.

4. Cherish, P., TREC Technical Report 61-47, Basic Curtiss-Wright Series 300-ST Turboshaft Engine Calibration, U.S. Army Transportation Research Command, Fort Eustis, Virginia, April 1961.

5. Technical Data, Model 300-ST Industrial Gas Turbine, Curtiss-Wright Corporation, Utica Division, Utica, Michigan, reprinted May 1958.

6. Liquid-Metals Handbook, Sodium-NaK Supplement, TID 5277, Atomic Energy Commission, Department of the Navy, Washington, D.C., 1 July 1955.

Figures

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

SPECIFIC FUEL CONSUMPTION OF CONFIGURATIONS USING DOWN STREAM HEAT EXCHANGERS BEFORE AND BEHIND POWER TURBINE

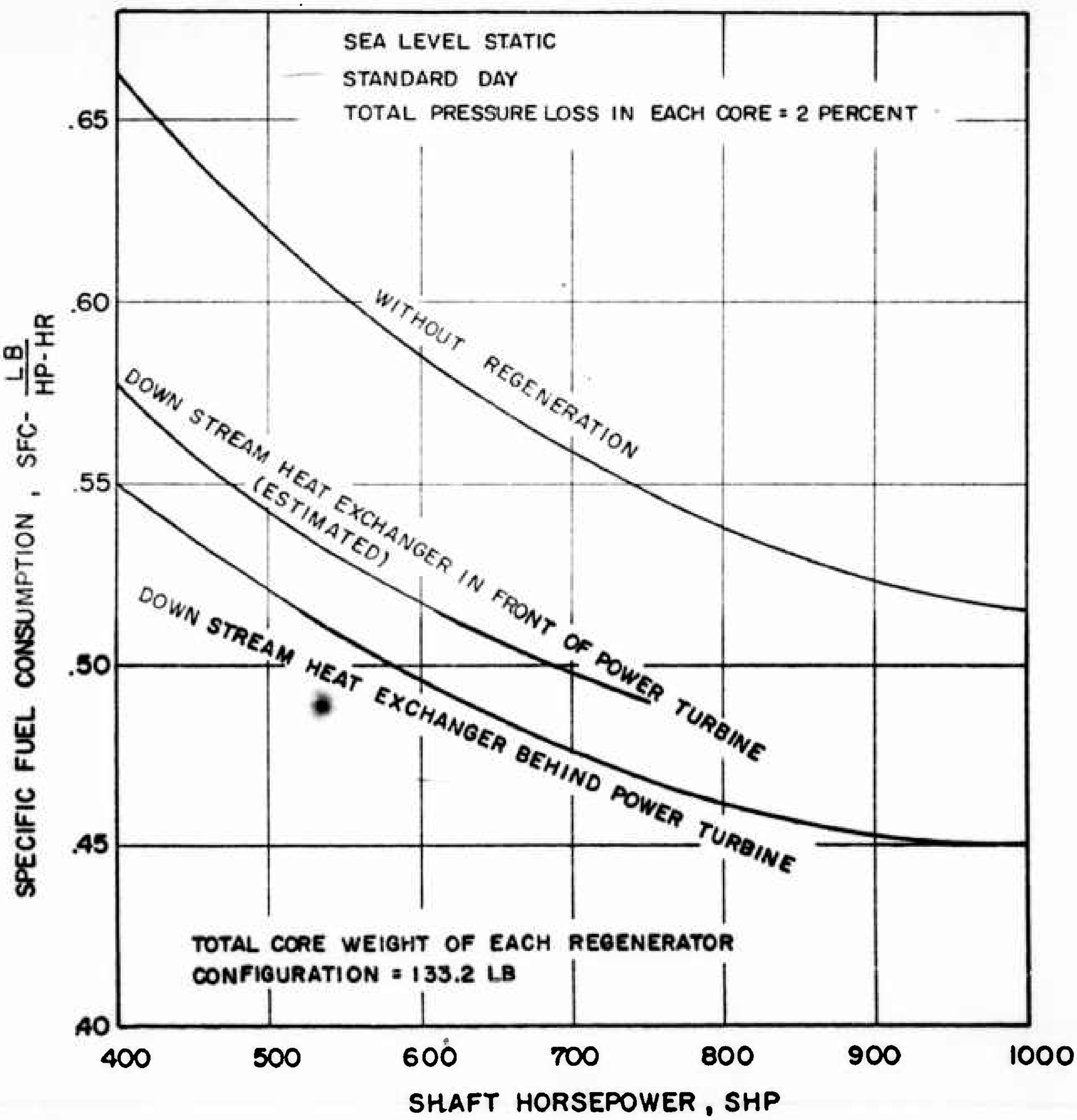


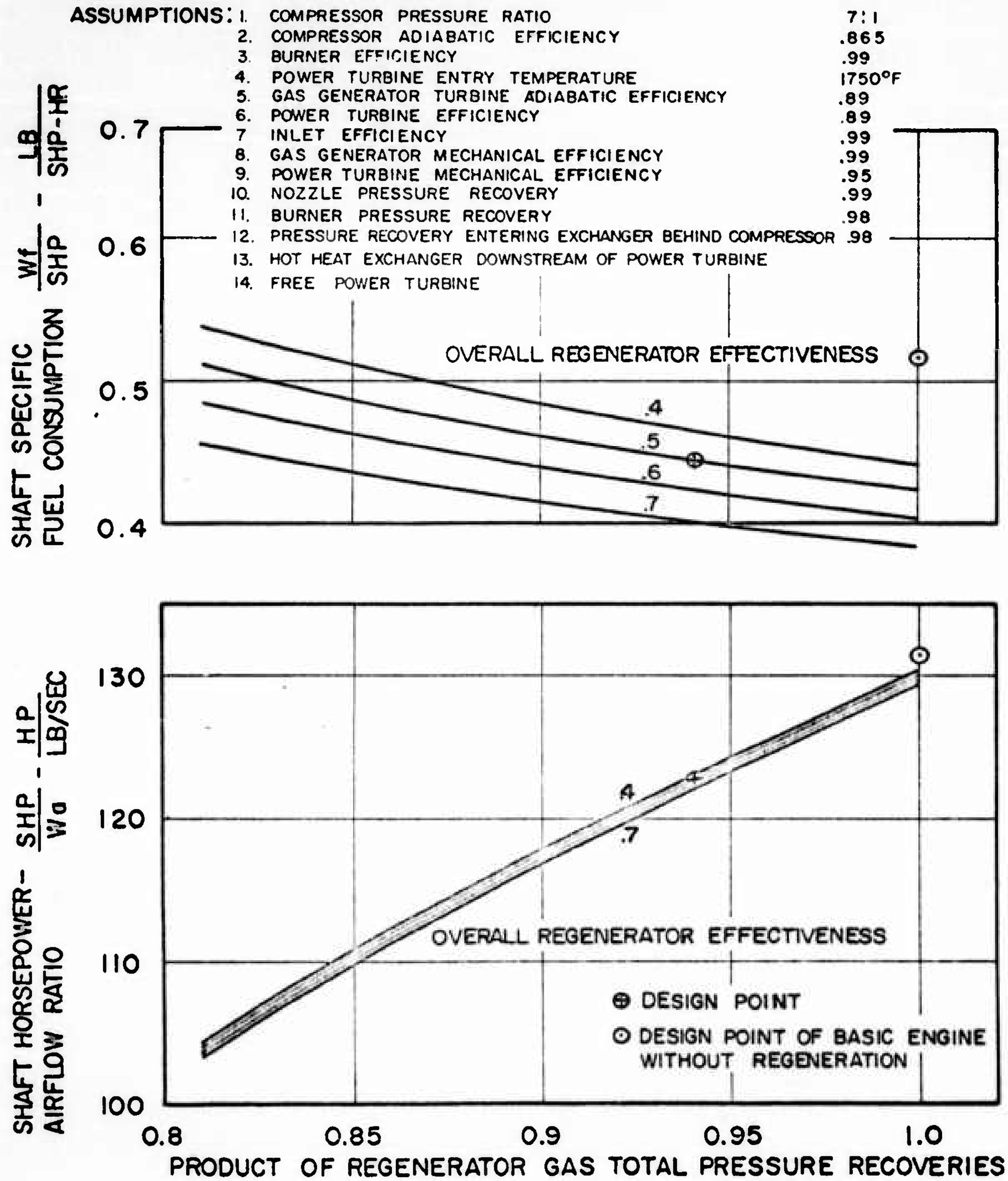
Figure 1

LIQUID METAL REGENERATOR FEASIBILITY STUDY

1000 HORSEPOWER TURBOSHAFT ENGINE

EFFECT OF REGENERATOR EFFECTIVENESS AND TOTAL PRESSURE RECOVERY ON ENGINE PERFORMANCE

DESIGN POINT DATA, SEA LEVEL STATIC, STANDARD DAY CONDITIONS



$$\left(1-\frac{\Delta P}{P}\right)_{\text{IN CORE BEHIND COMPRESSOR}} \times \left(1-\frac{\Delta P}{P}\right)_{\text{IN CORE BEHIND TURBINE}} \times \left(1-\frac{\Delta P}{P}\right)_{\text{ENTERING EXCHANGER BEHIND TURBINE}}$$

Figure 2

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

ESTIMATED EFFECT OF DESIGN POINT TURBINE ENTRY TEMPERATURE
ON
SPECIFIC FUEL CONSUMPTION
FOR
7:1 COMPRESSOR PRESSURE RATIO
AND
.5 OVERALL REGENERATOR EFFECTIVENESS

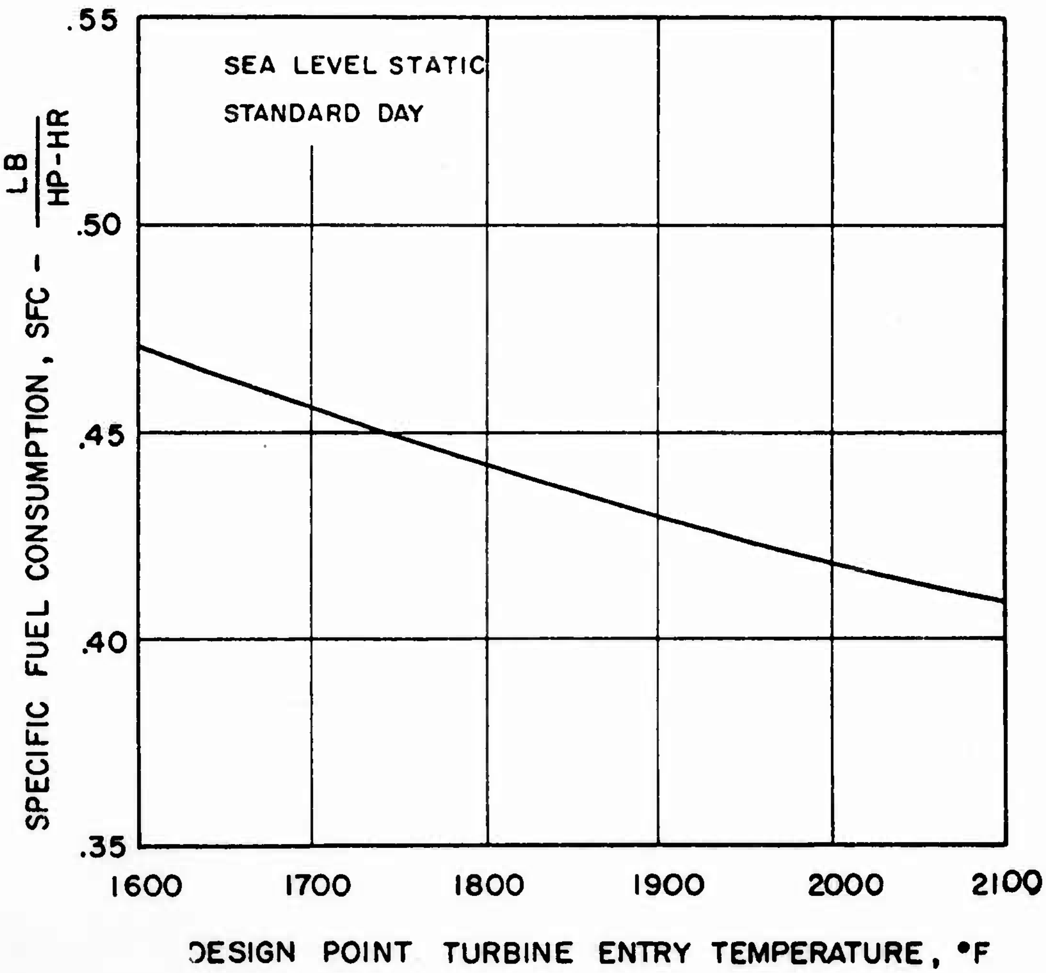


Figure 3

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

ESTIMATED EFFECT OF DESIGN POINT TURBINE ENTRY TEMPERTURE
ON
SPECIFIC SHAFT HORSEPOWER
FOR
7:1 COMPRESSOR PRESSURE RATIO
AND
.5 OVERALL REGENERATOR EFFECTIVENESS

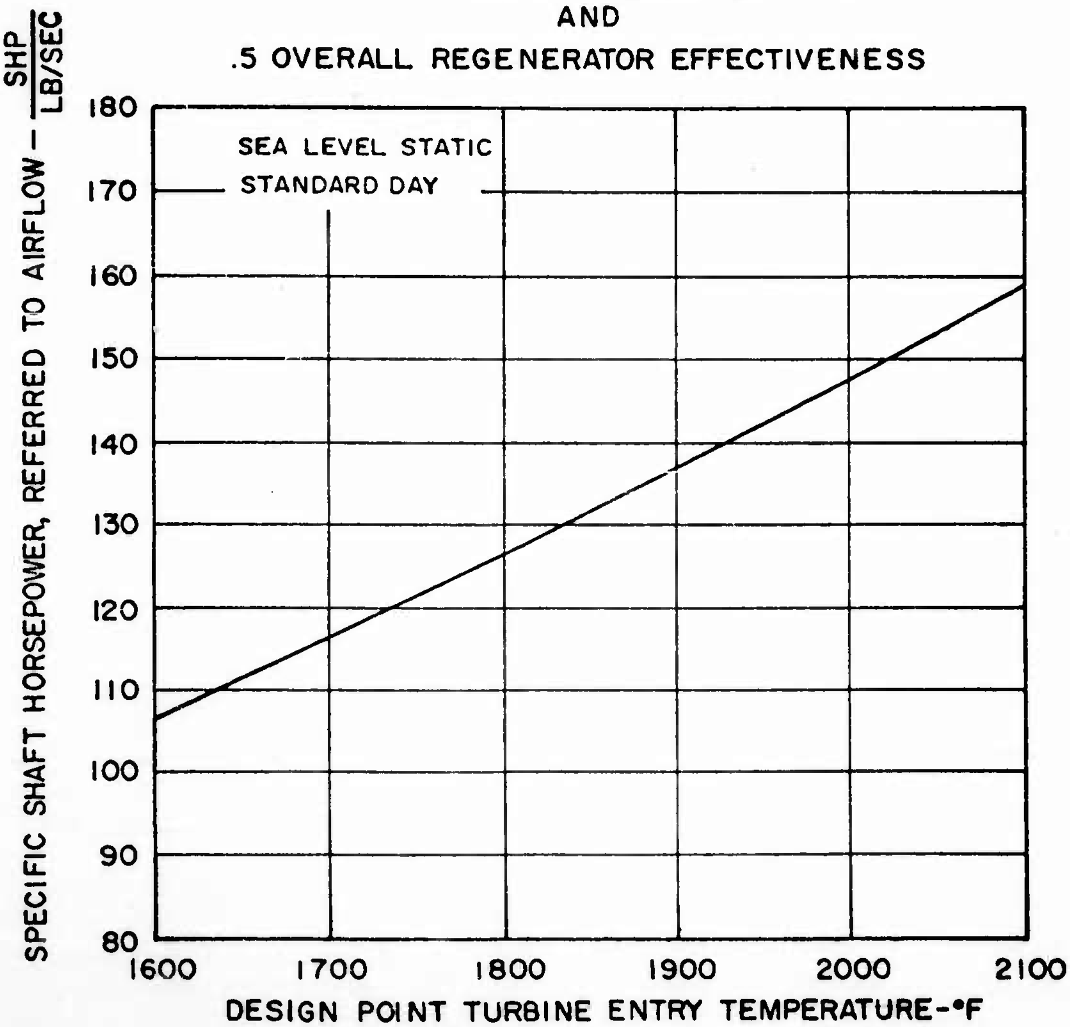


Figure 4

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

ESTIMATED EFFECT OF DESIGN POINT COMPRESSOR TOTAL PRESSURE RATIO ON
SPECIFIC FUEL CONSUMPTION
FOR
1750°F TURBINE ENTRY TEMPERATURE
AND
.5 OVERALL REGENERATOR EFFECTIVENESS

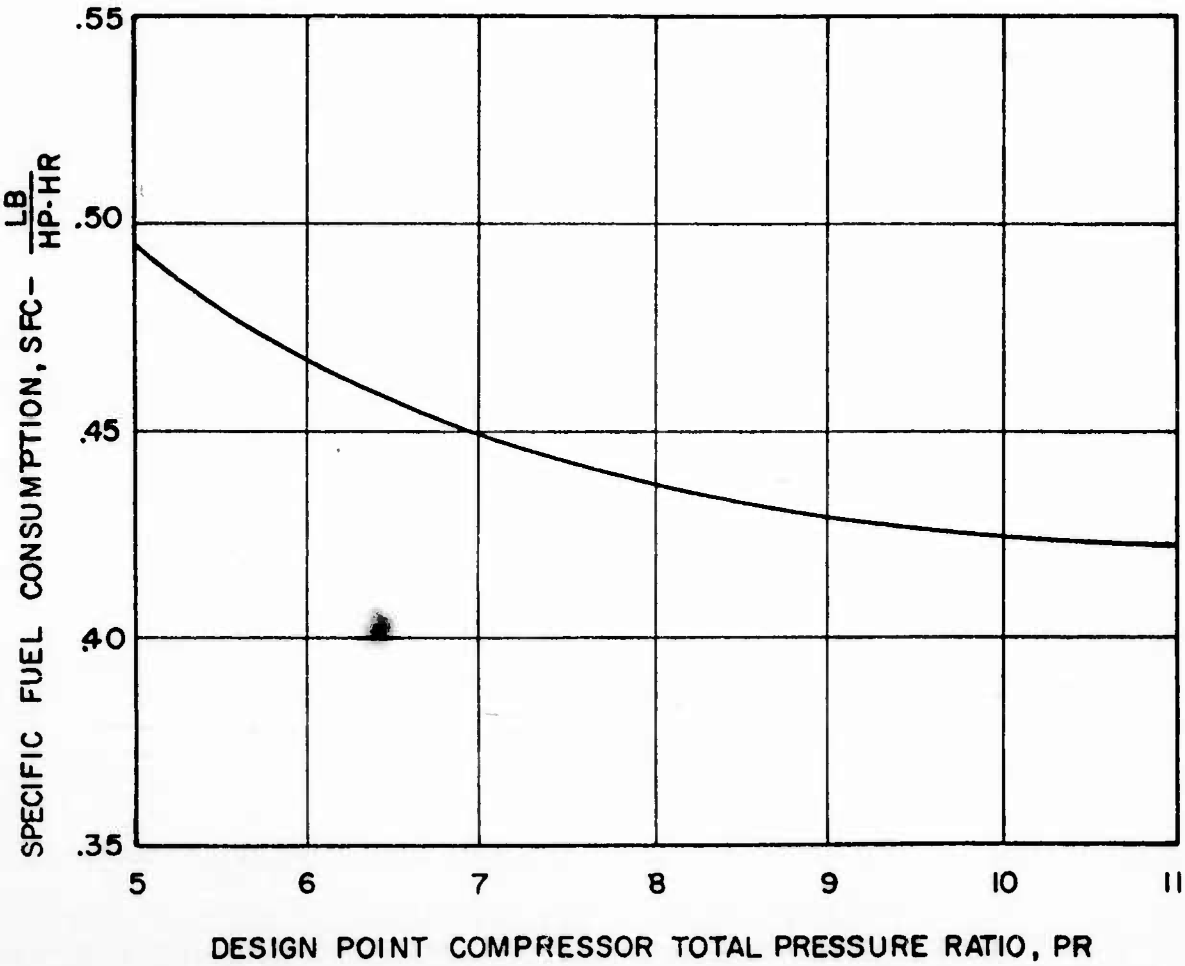


Figure 5

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

ESTIMATED EFFECT OF DESIGN POINT COMPRESSOR TOTAL PRESSURE RATIO ON SPECIFIC SHAFT HORSEPOWER
FOR
1750°F TURBINE ENTRY TEMPERATURE
AND
.5 OVERALL REGENERATOR EFFECTIVENESS

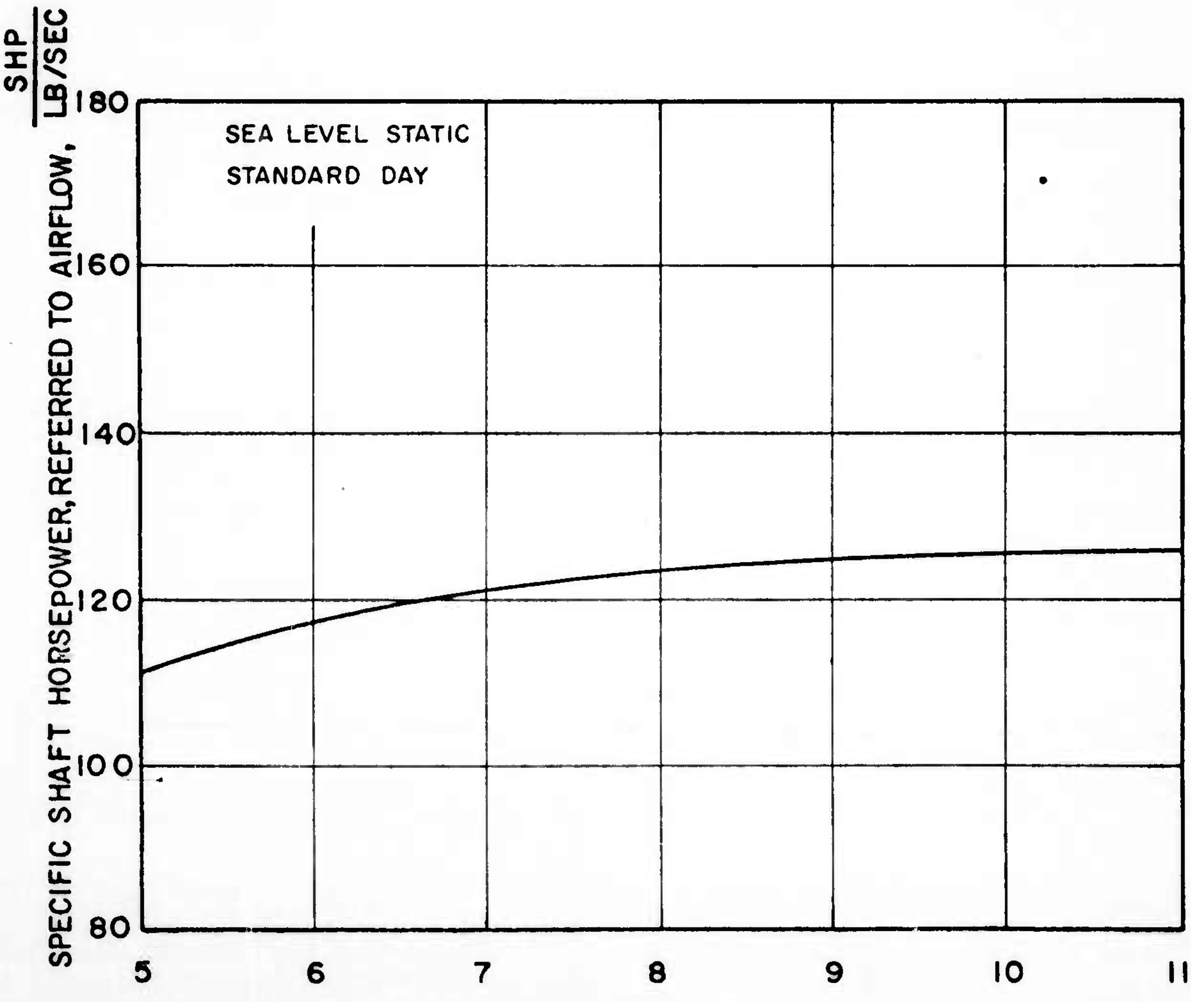


Figure 6

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

SPECIFIC FUEL CONSUMPTION VERSUS THE
ENGINE ROTATIONAL SPEEDS

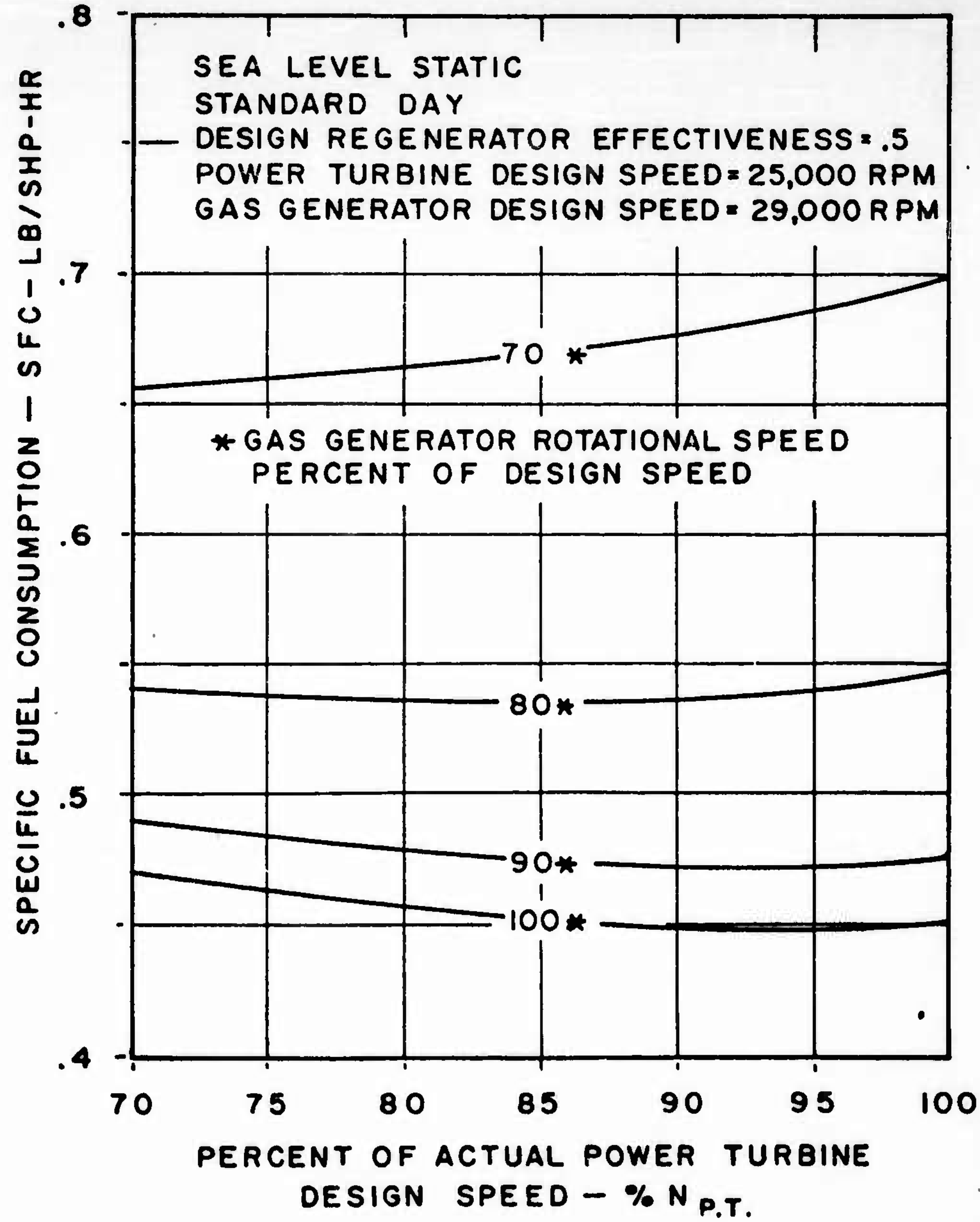


Figure 7

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

SHAFT HORSEPOWER VERSUS THE ENGINE ROTATIONAL SPEEDS

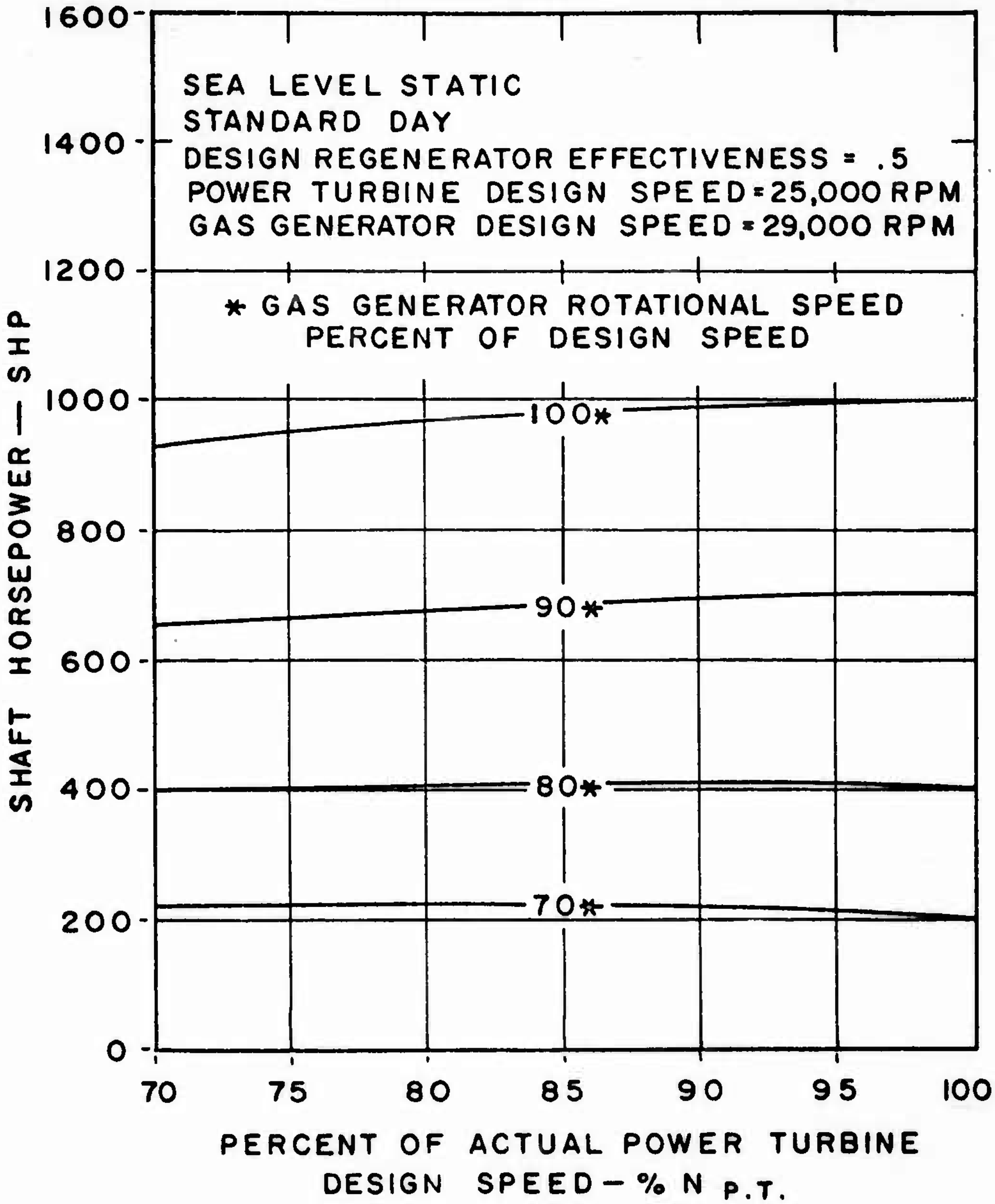


Figure 8

LIQUID METAL REGENERATOR FEASIBILITY STUDY

1000 HORSEPOWER TURBOSHAFT ENGINE

ENGINE PERFORMANCE CURVES, ROTATIONAL SPEEDS, TURBINE ENTRY TEMPERATURE, EXHAUST GAS TEMPERATURE AND SPECIFIC FUEL CONSUMPTION vs SHAFT HORSEPOWER

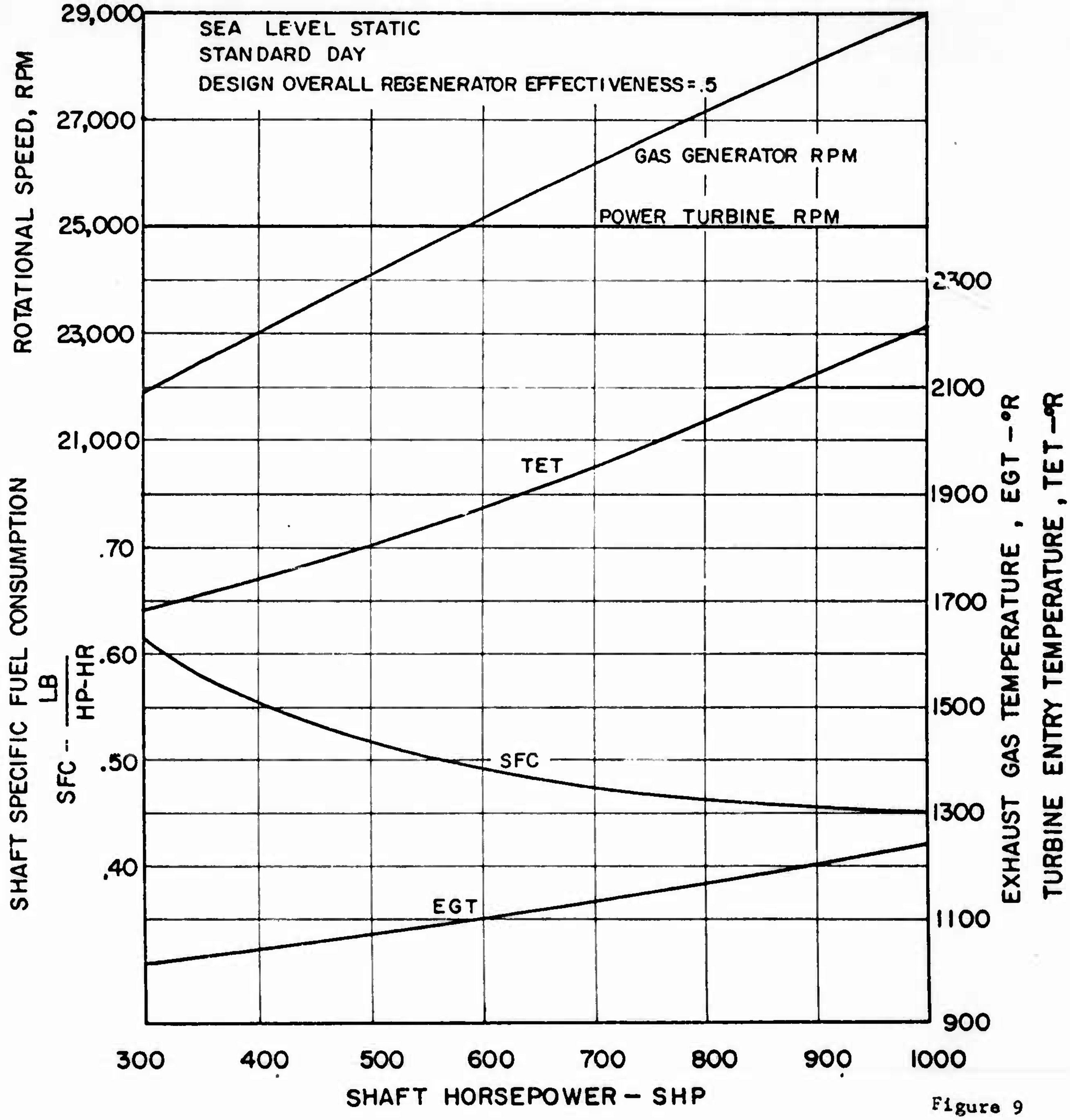


Figure 9

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

ENGINE PERFORMANCE CURVES- FUEL FLOW, AIR WEIGHT FLOW, COMPRESSOR PRESSURE RATIO AND OVERALL REGENERATOR EFFECTIVENESS vs SHAFT HORSEPOWER

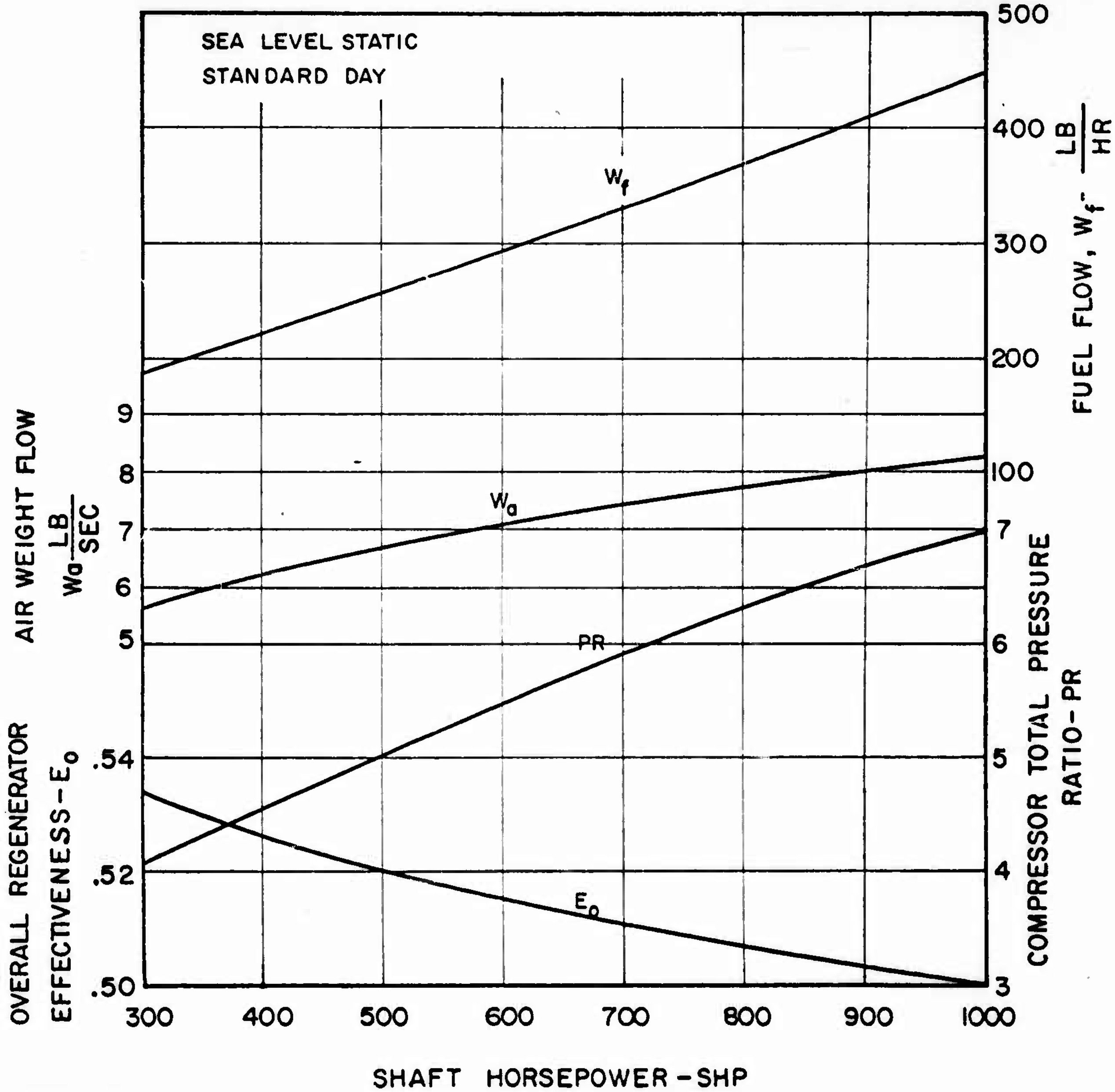


Figure 10

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

HEAT EXCHANGER PERFORMANCE CURVES-TEMPERATURES OF GAS AND NAK ENTERING AND LEAVING, OVERALL REGENERATOR EFFECTIVENESS vs SHAFT HORSEPOWER

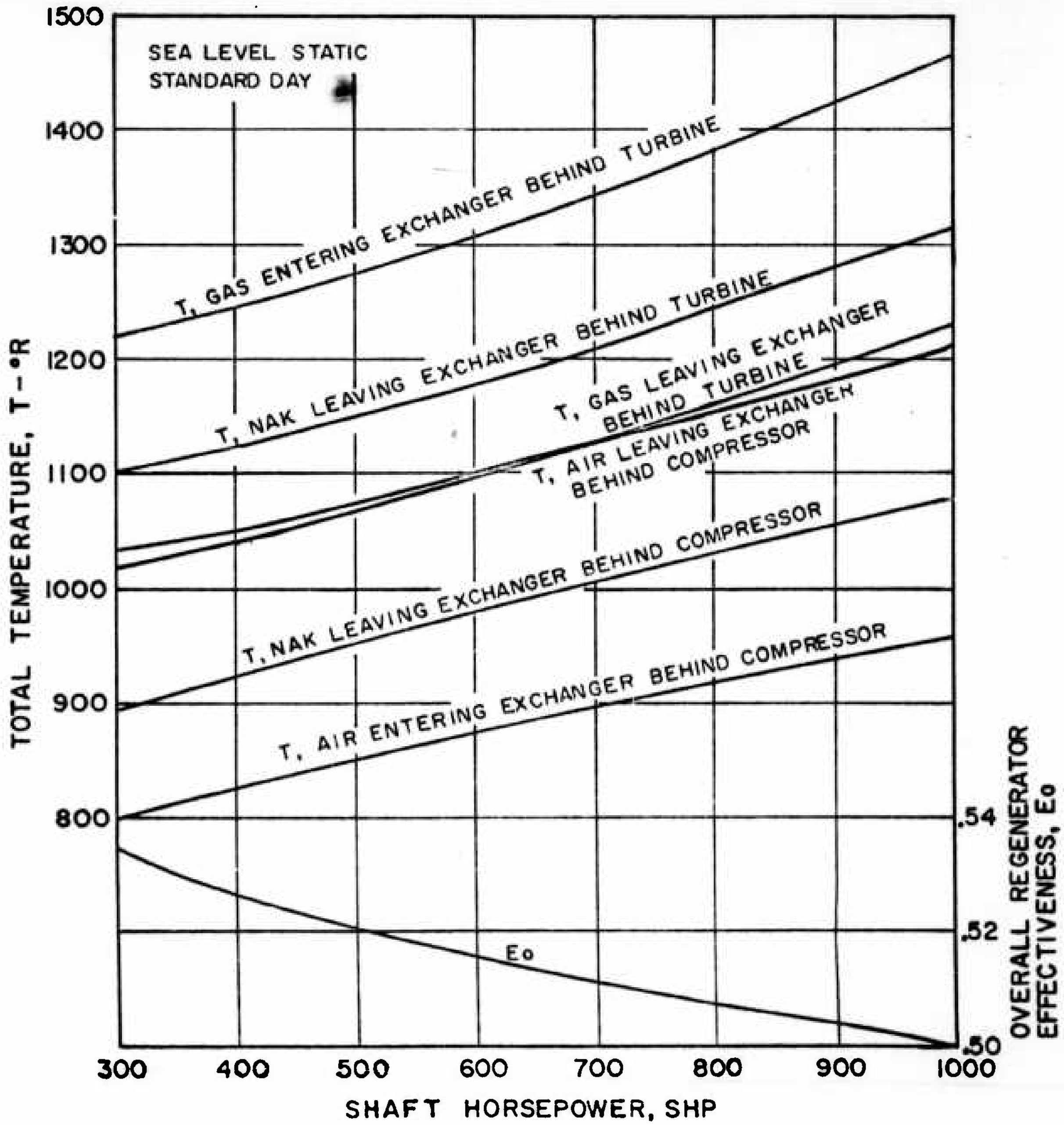


Figure 11

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

HEAT EXCHANGER PERFORMANCE CURVES - ENTERING GAS VELOCITY, TOTAL PRESSURE LOSS (ENTERING + CORE) AND ENTERING TOTAL PRESSURE vs SHAFT HORSEPOWER

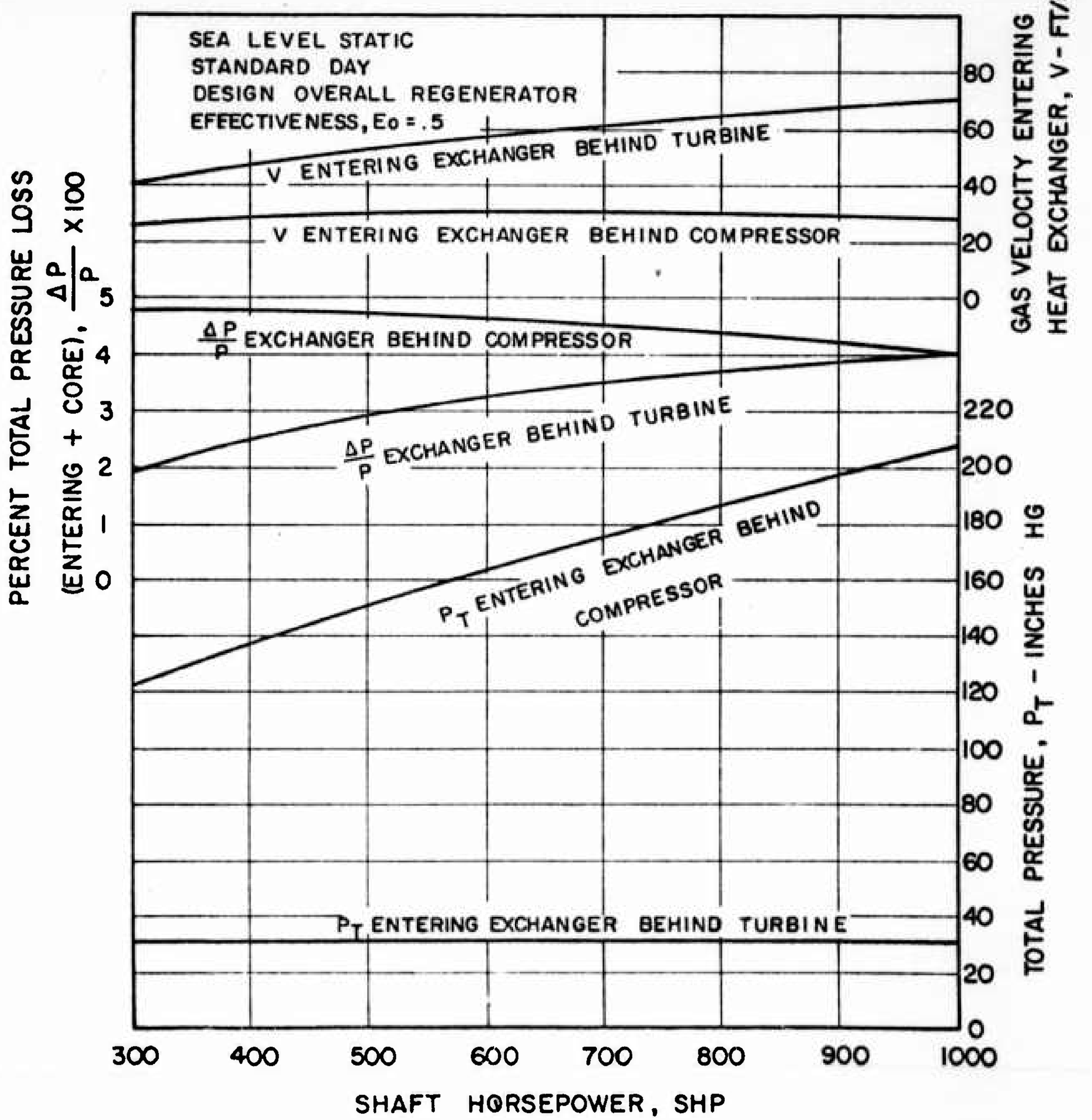


Figure 12

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

COMPRESSOR MAP SHOWING
OPERATING LINE

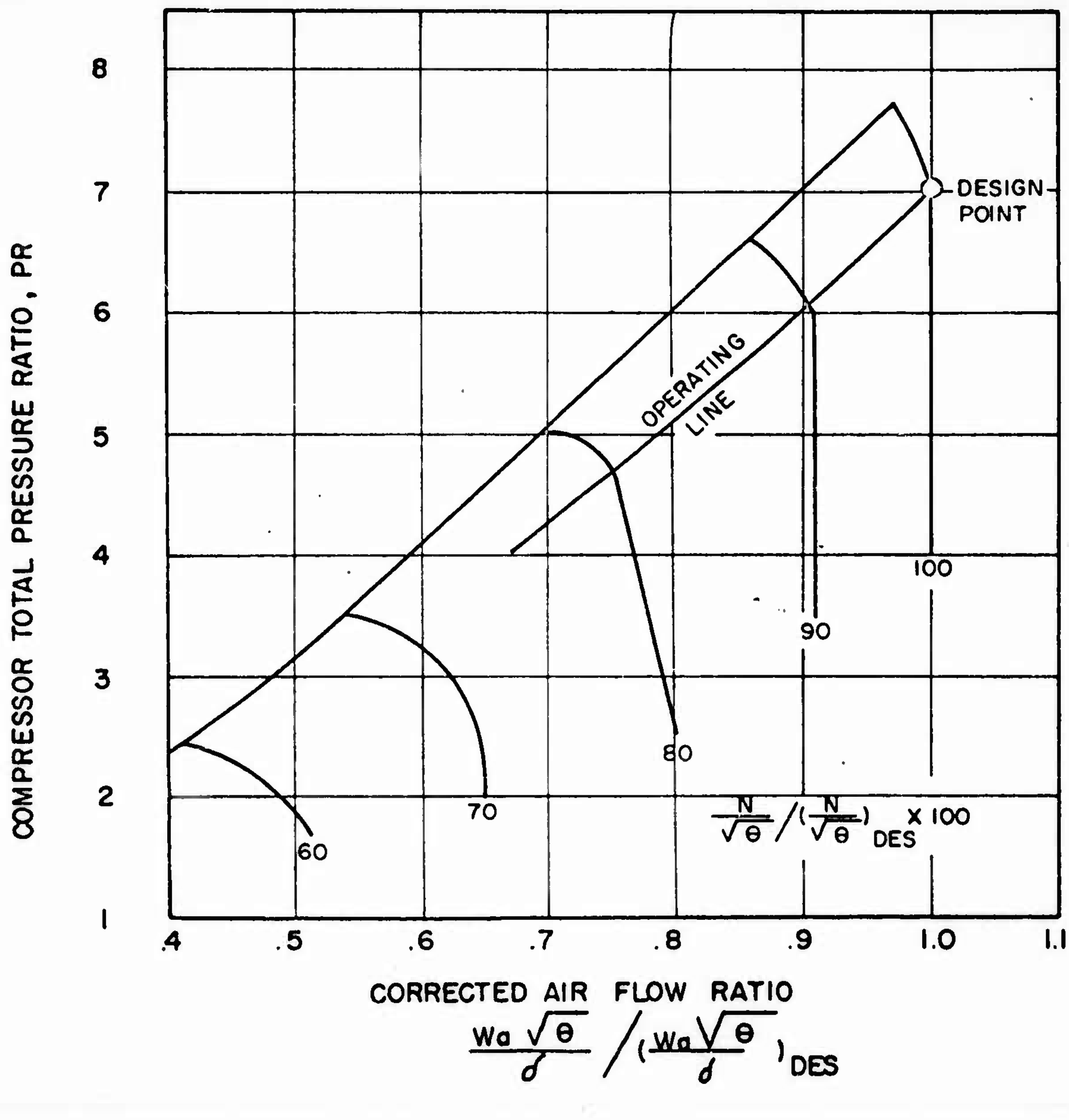


Figure 13

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

COMPONENT REGENERATOR EFFECTIVENESS RATIO
vs
TOTAL WEIGHT OF BOTH CORES
FOR
VARIOUS VALUES OF OVERALL REGENERATOR EFFECTIVENESS

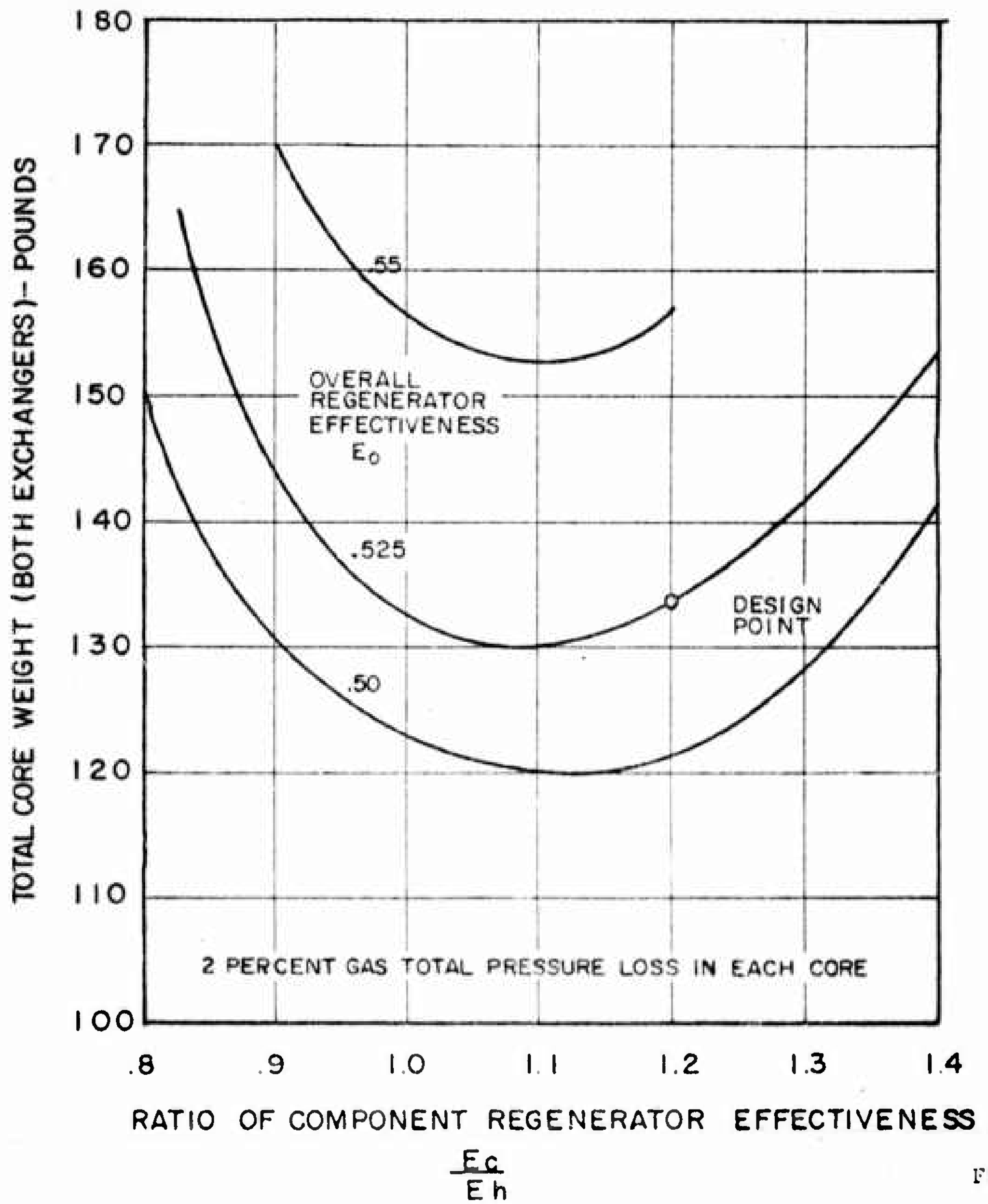


Figure 14

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE WEIGHTS INCLUDING NAK

vs

CORE FACE AREA

FOR

VARIOUS NUMBERS OF TUBE LAYERS

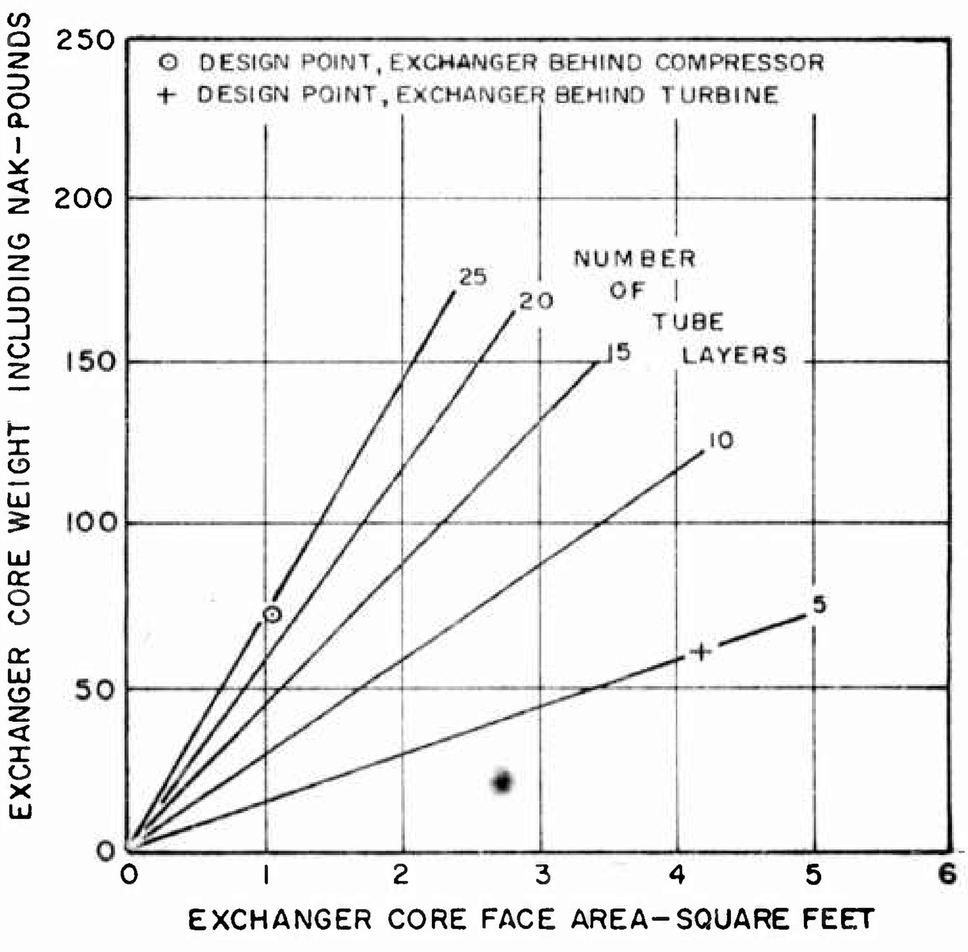


Figure 15

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

COMPONENT REGENERATOR EFFECTIVENESS
vs
EXCHANGER CORE FACE AREA
FOR
EXCHANGER BEHIND COMPRESSOR AND
EXCHANGER BEHIND TURBINE

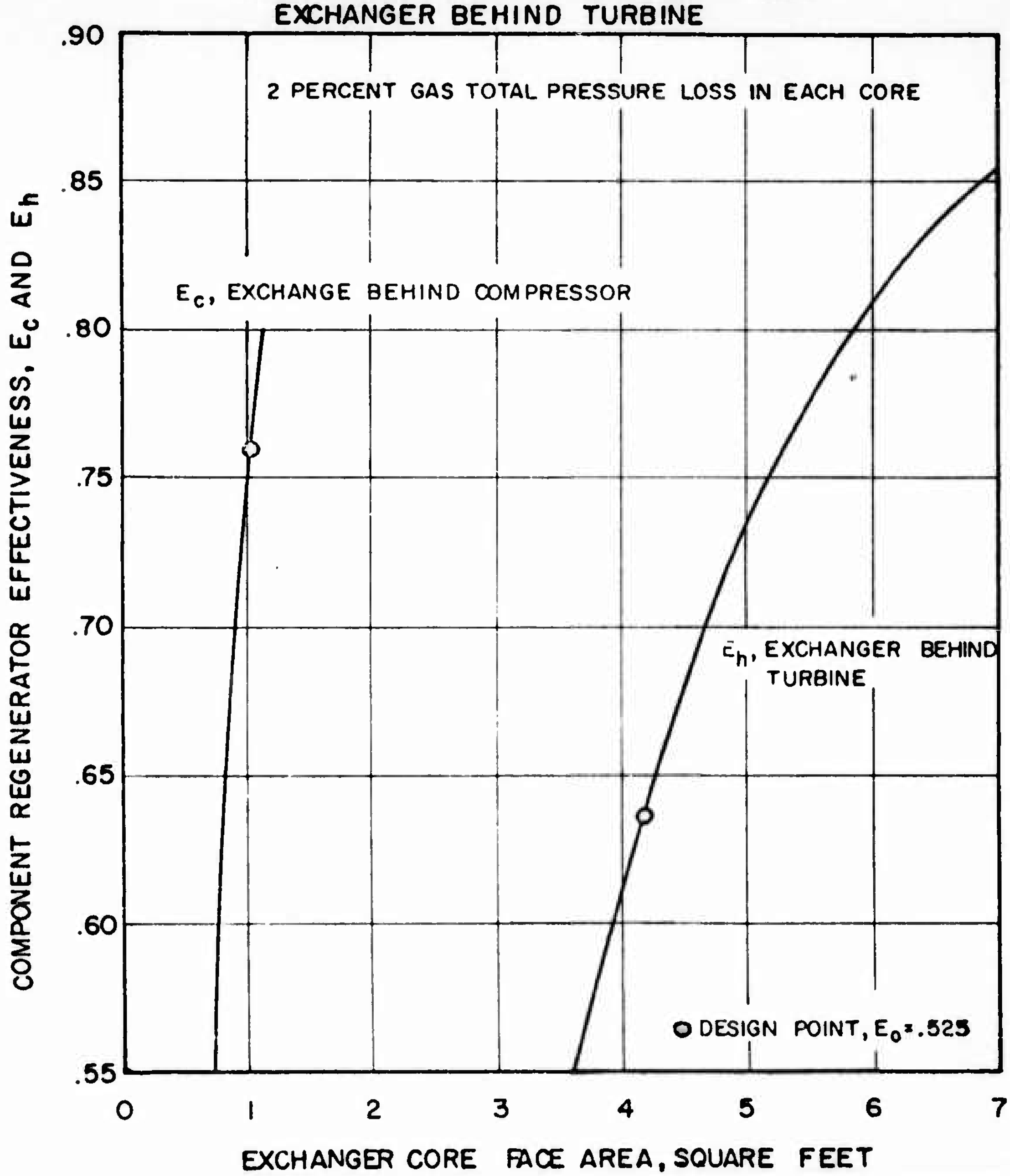


Figure 16

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE
HEAT EXCHANGER CORE WEIGHT
VS
COMPONENT REGENERATOR EFFECTIVENESS
FOR
EXCHANGER BEHIND COMPRESSOR AND
EXCHANGER BEHIND TURBINE

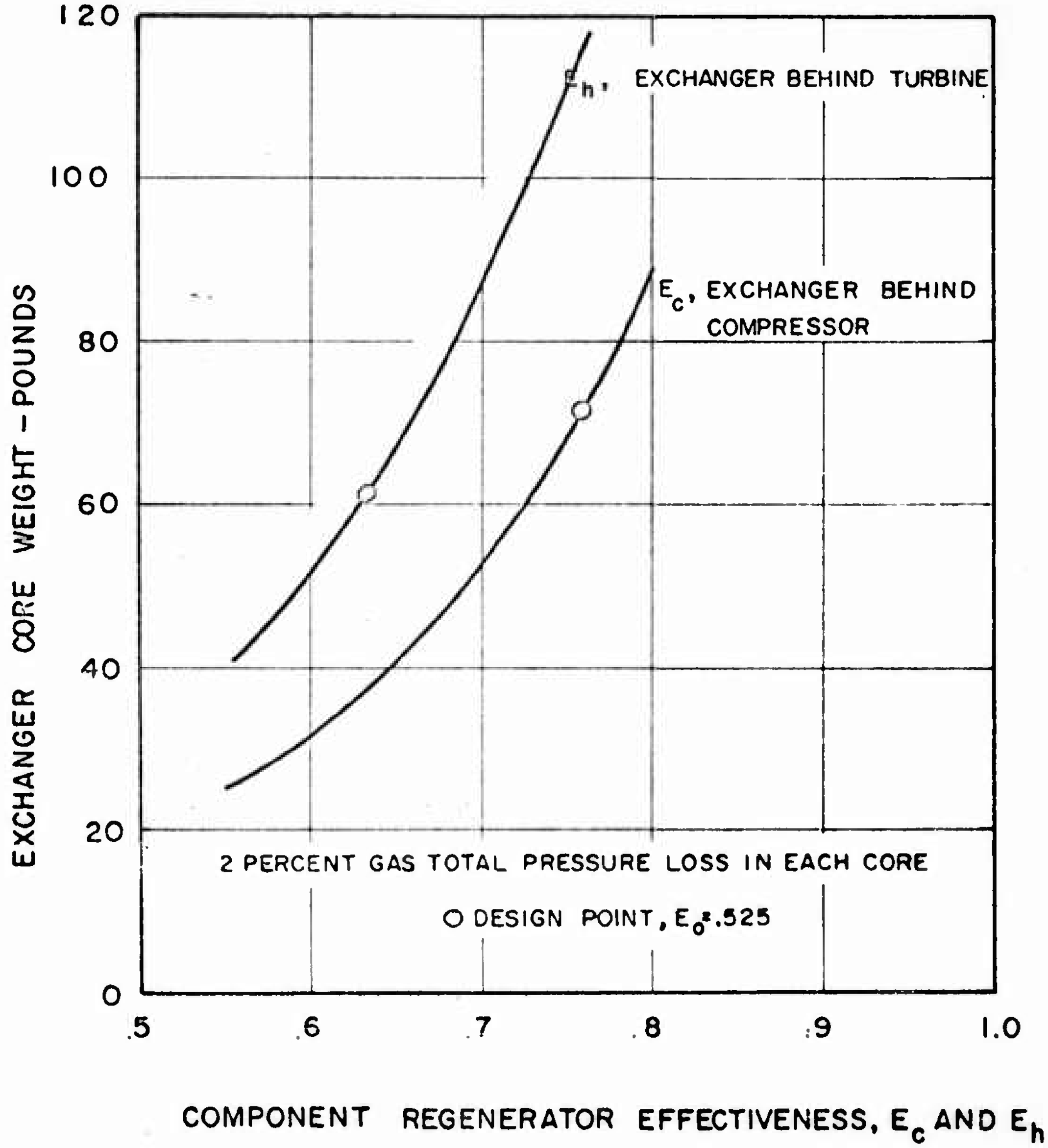


Figure 17

Appendix A
1000-Horsepower Engine Heat Exchangers Parametric Design Study

In the design of extended surface heat exchangers for application to gas turbine regenerators there are a number of variables to which values have to be given. In an attempt to determine the effect of these parameters on heat exchanger core weight and face area, and on the number of tube layers required, an extensive study was made in which many exchangers were parametrically designed for a 1000-horsepower turboshaft engine having a 7/1 compressor pressure ratio and a 1750°F turbine entry temperature. Core weight, face area and number of tube layers were determined for various overall regenerator effectivenesses (E_o), percent total pressure loss, ($\frac{\Delta P}{P}$), and number of fins per inch. This calculation was done for the two fin configurations outlined in the table on page 78. Fin Configuration 1 is this company's design and is the main object of this study. For comparison, however, Fin Configuration 2 (a standard production finned tube) was analized for one overall regenerator effectiveness. In addition, a computation was made showing the effect of fin thickness for one set of fixed values of E_o, $\frac{\Delta P}{P}$, and fins per inch.

The above computations were based on the assumption that the heat exchangers behind the compressor and behind the turbine were of equal effectiveness. A computation was then made to determine the influence of effectiveness ratios other than unity on the combined weight of both cores.

The calculation procedure paralled the design point calculation procedure presented in this report on page 32. Deviations from this procedure were as follows:

1. The NaK film heat transfer coefficients (h_L) were computed by a method presented in Reference 6, page 73, equation 1, giving values between 13,000 and 14,000 compared to the constant 15,660 BTU per hour per square feet per °F used in the design point calculation.

2. No fouling factor is considered.

ENGINE PERFORMANCE USED FOR THIS STUDY

T_{ah1} = 1532°R

T_{ac1} = 959°R

P_{ah1} = 2176 lb/ft^2

P_{ac1} = 14520 lb/ft^2

W_a = 8.25 lb/sec

REGENERATOR ASSUMPTIONS

$$C_{ah} = C_{ac} = C_L$$

$$\overline{T}_{ah} = T_{ah1} - E_o \frac{(T_{ah1} - T_{ac1})}{2}$$

$$\overline{T}_{ac} = T_{ac1} + E_o \frac{(T_{ah1} - T_{ac1})}{2}$$

DEFINITION OF SYMBOLS USED

C	Fluid heat capacity = WC_p
C_p	Specific heat at constant pressure, $\frac{BTU}{lb-^oF}$
E_c	Regenerator effectiveness of exchanger behind compressor $= \frac{T_{ac2}-T_{ac1}}{T_{L2}-T_{ac1}}$
E_h	Regenerator effectiveness of exchanger behind turbine $= \frac{T_{L2}-T_{L1}}{T_{ah1}-T_{L1}}$
E_o	Overall regenerator effectiveness $= \frac{T_{ac2}-T_{ac1}}{T_{ah1}-T_{ac1}}$
P	Total pressure, lb/ft^2
T	Total temperature, ^oR
$\overline{T}$	Mean Temperature, ^oR
W	Weight flow, lb/sec

SUBSCRIPTS

a	Air or gas
c	Heat exchanger behind compressor
h	Heat exchanger behind turbine
L	Liquid NaK
o	Overall regenerator
1	Entering
2	Leaving

(As applied to the liquid NaK relative to the heat exchangers, "leaving" and "entering" refers to the heat exchanger behind the turbine)

TABLE: FINNED TUBE CONFIGURATIONS

All Dimensions in Inches

	CONFIGURATION 1	CONFIGURATION 2	1000-HP Engine Design (for comparison only, not included in this study)
Outside diameter of fin	.418	.375	.418
Outside diameter of tube	.15625	.1875	.15625
Thickness of copper	.008	.004	.004
Thickness of clad	.001	.002	.002
Total fin thickness	.010	.008	.008
Thickness of fin skirt	.0035	none	none
Thickness of tube wall	.015	.020	.015

DISCUSSION OF RESULTS

The results obtained using fin Configuration 1 and 2 with constant fin thickness and constant effectiveness ratios are presented in Figures a through x where the variation of individual core pressure loss, exchanger face area, and core weights versus number of tube layers are shown for 20, 24.4 and 30 fins per inch, .50, .55, and .60 overall effectiveness and percent total pressure losses in the cores of 1, 2, 3 and 4. In addition, total core weight (both exchangers) is presented versus overall regenerator effectiveness and number of fins per inch for the same values of Eo, fins per inch and percent of total pressure loss as above.

The following general conclusions are based on these data:

1. Reduction of total pressure losses in the cores below 2% of entering pressure involves rapid increases in required face areas and resulting weights (see Figures g, i, and o-x).

2. Exchanger face area increases as the number of fins per inch increases but core weight, which is, in addition, a function of the decreasing number of tube layers, decreases continuously up to at least 30 fins per inch.* (see Figures d-i and m-x).

3. The higher overall regenerator effectiveness require greater face areas and core weights for the same pressure loss and fin configuration. (see Figures d-i and m-x).

4. The exchanger behind the turbine need incorporate considerably fewer tube layers for the same effectiveness and pressure loss as the exchanger behind the compressor but requires considerably greater face area and twice as much weight (for example, compare Figures a and j, d and m, also g and p).

5. Fin-tube Configuration 2 yields core weights which are somewhat less than those obtained with Configuration 1 and face areas and numbers of tube layers which are somewhat higher. Apparently the effects of the deletion of the fin skirt and the reduction of the fin thickness and diameter outweigh the effects of increasing tube size and wall thickness. No attempt is made here to isolate these functions (see Figures a, d, g, j, o, p, and u).

On Figures y through bb are presented the effects of varying the thickness of the copper in the fins on number of tube layers, exchanger face area, and core weight. It can be seen that both face area and core weight decrease with copper thickness to values of approximately .003 inches.

*There is a practical limit to the number of fins per inch which would continue to give reduced weights in that the fin efficiency would be adversely affected when the fin spacing became sufficiently small to cause interference of the boundary layers on each surface.

Also, on these same figures (y through bb) the results obtained when a solid aluminum fin is used in the exchanger behind the compressor are presented for comparison. It can be seen that the core with the aluminum fin is materially lighter (Figure y) despite increased requirement for number of tube layers (Figure z). This increase in tube layers is a function of the lower thermal conductivity of the aluminum. Core weight, using aluminum fins, does not increase with fin thickness as rapidly as when using clad copper fins.

It should be noted that the parameters for the stainless-steel-clad fins are plotted against the copper thickness only (total thickness less .002 inches), whereas for the aluminum the thickness is plotted against the total thickness. Except for the calculation of fin efficiency, as mentioned above, the effect of the stainless steel cladding was taken fully into account. Plotted against the copper thickness only, the heat transfer function of the thickness is emphasized and, in particular, for the comparison with the aluminum.

The face areas obtained utilizing copper and aluminum fins are presented on Figure bb. Here it can be seen that for all values of thickness greater than optimum, the aluminum fin appears to require lower face areas than the clad copper. If the curve for the clad copper fin is adjusted for cladding thickness, the two curves are practically the same.

On Figure cc total core weight is shown versus effectiveness ratio using a constant overall effectiveness of .55, .24.4 fins per inch of the fin Configuration 1, and a core total pressure loss of 2 percent in each exchanger. It can be seen that the optimum component regenerator effectiveness ratio for these conditions is approximately 1.10, the exchanger behind the compressor having the larger effectiveness. Since it can be shown elsewhere that at an overall ideal effectiveness of 1 the optimum effectiveness ratio likewise occurs at 1.0, it is concluded that at overall effectivenesses greater than .55 the optimum effectiveness ratio will be between 1.00 and 1.10 and that for overall effectiveness less than .55 the optimum effectiveness ratio will be greater than 1.10. It is significant to note the importance of maintaining any unbalance in the system in favor of higher component effectiveness in the exchanger behind the compressor since the alternative causes a rapid increase in overall core weight (see also Figure 14).

Figures

APPENDIX A

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE PRESSURE LOSS VS NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND COMPRESSOR

50 PERCENT OVERALL REGENERATOR EFFECTIVENESS

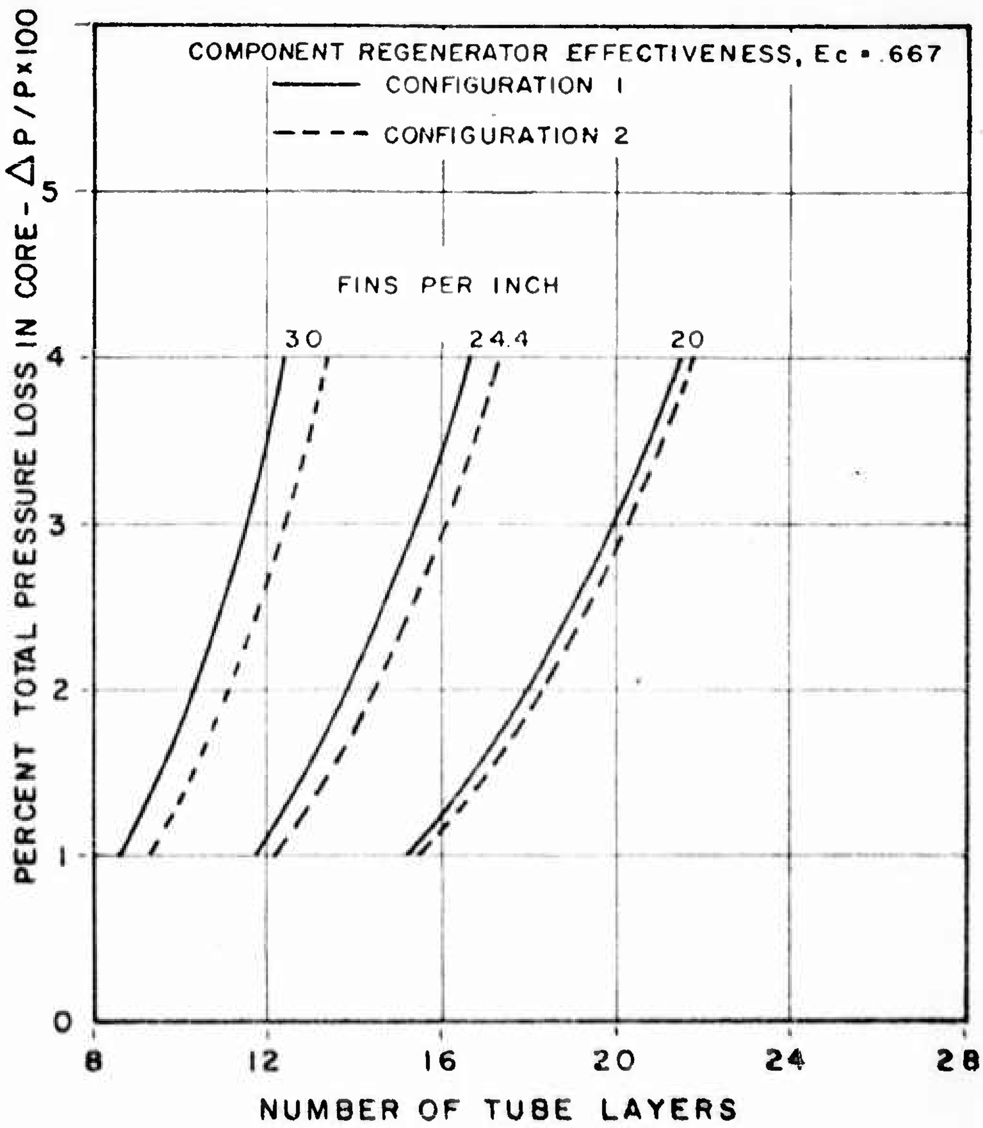


Figure a

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE TOTAL PRESSURE LOSS vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND COMPRESSOR

55 PERCENT OVERALL REGENERATOR EFFECTIVENESS

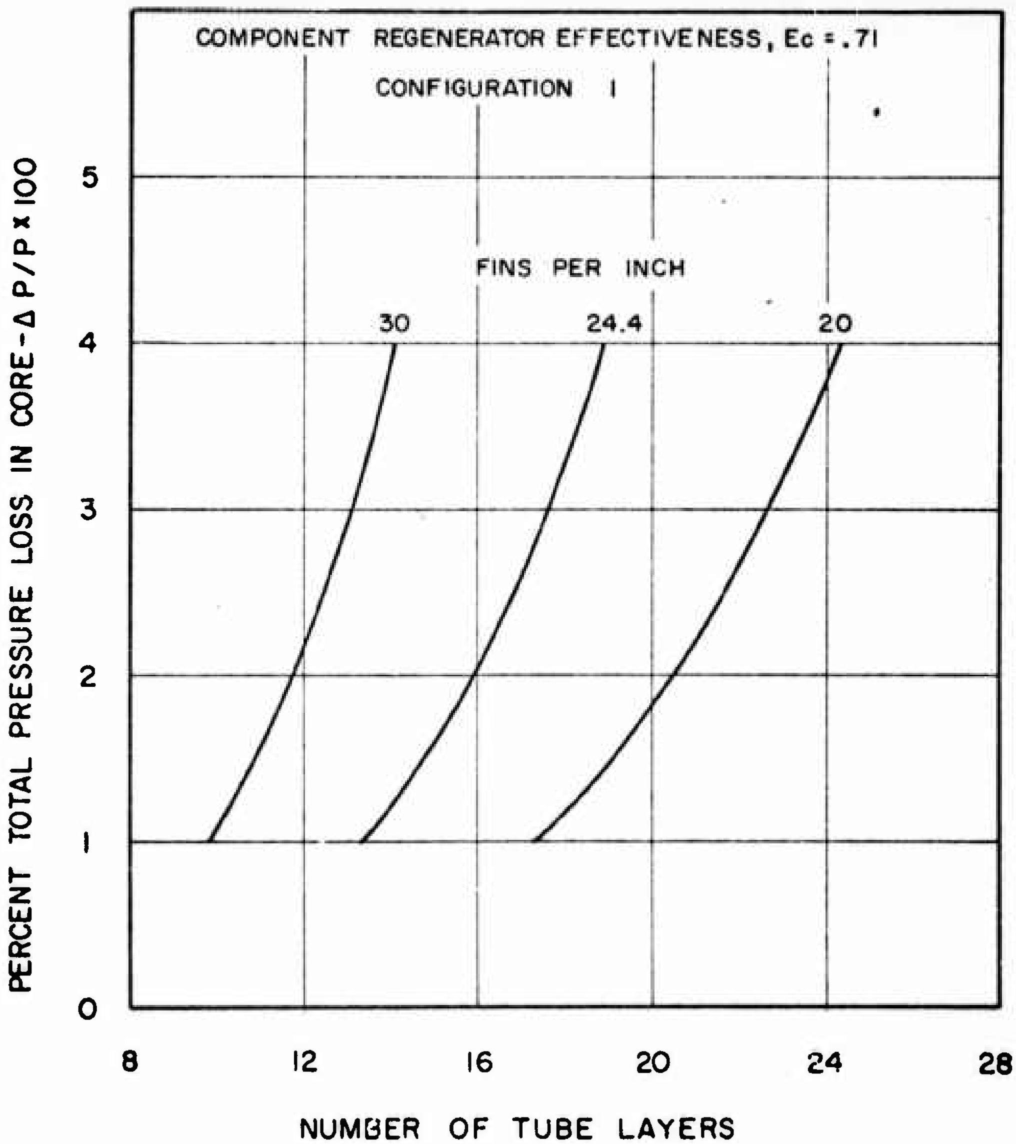


Figure b

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE TOTAL PRESSURE LOSS vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND COMPRESSOR

60 PERCENT OVERALL REGENERATOR EFFECTIVENESS

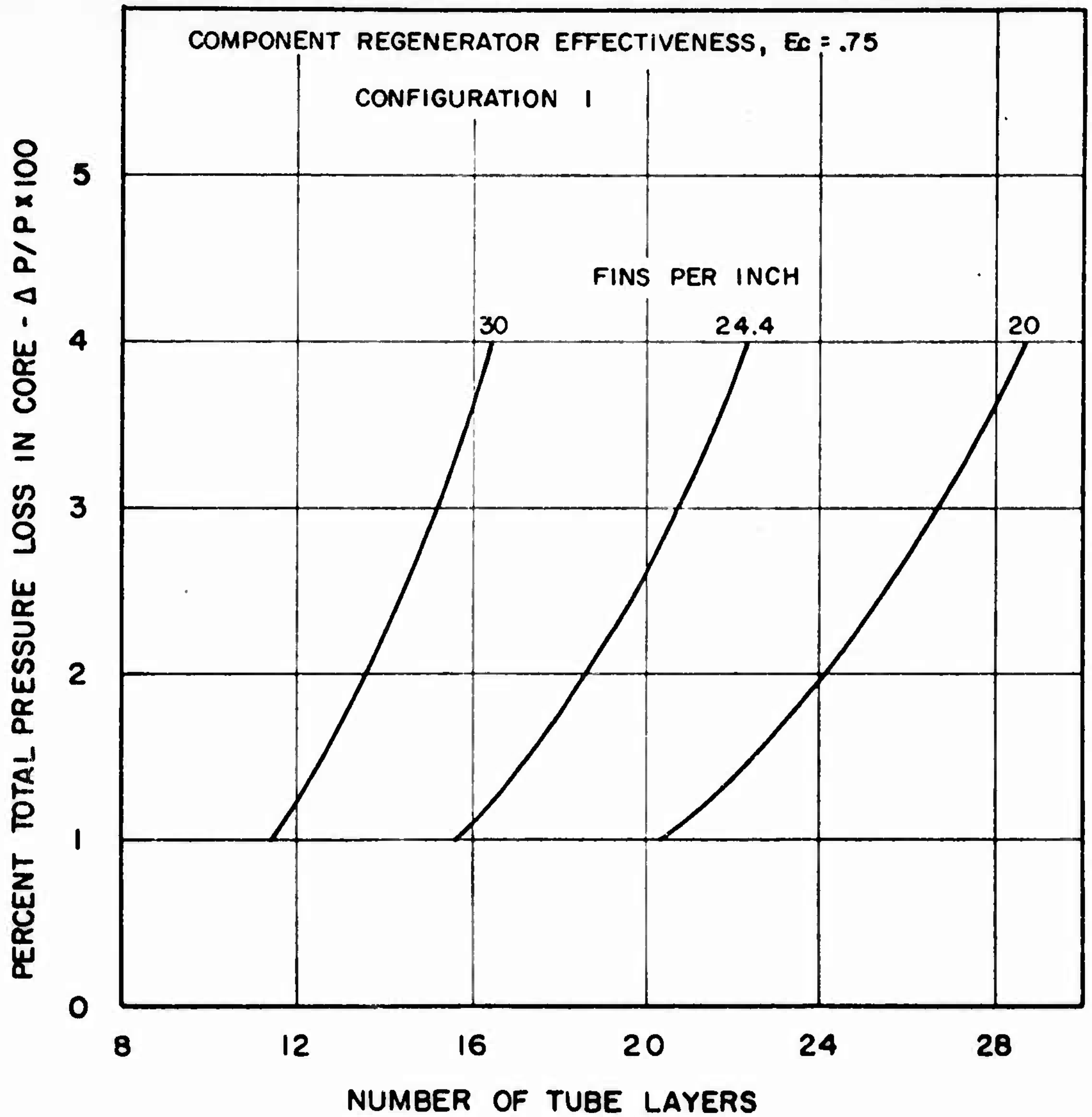


Figure c

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

FACE AREA vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND COMPRESSOR

50 PERCENT OVERALL REGENERATOR EFFECTIVENESS

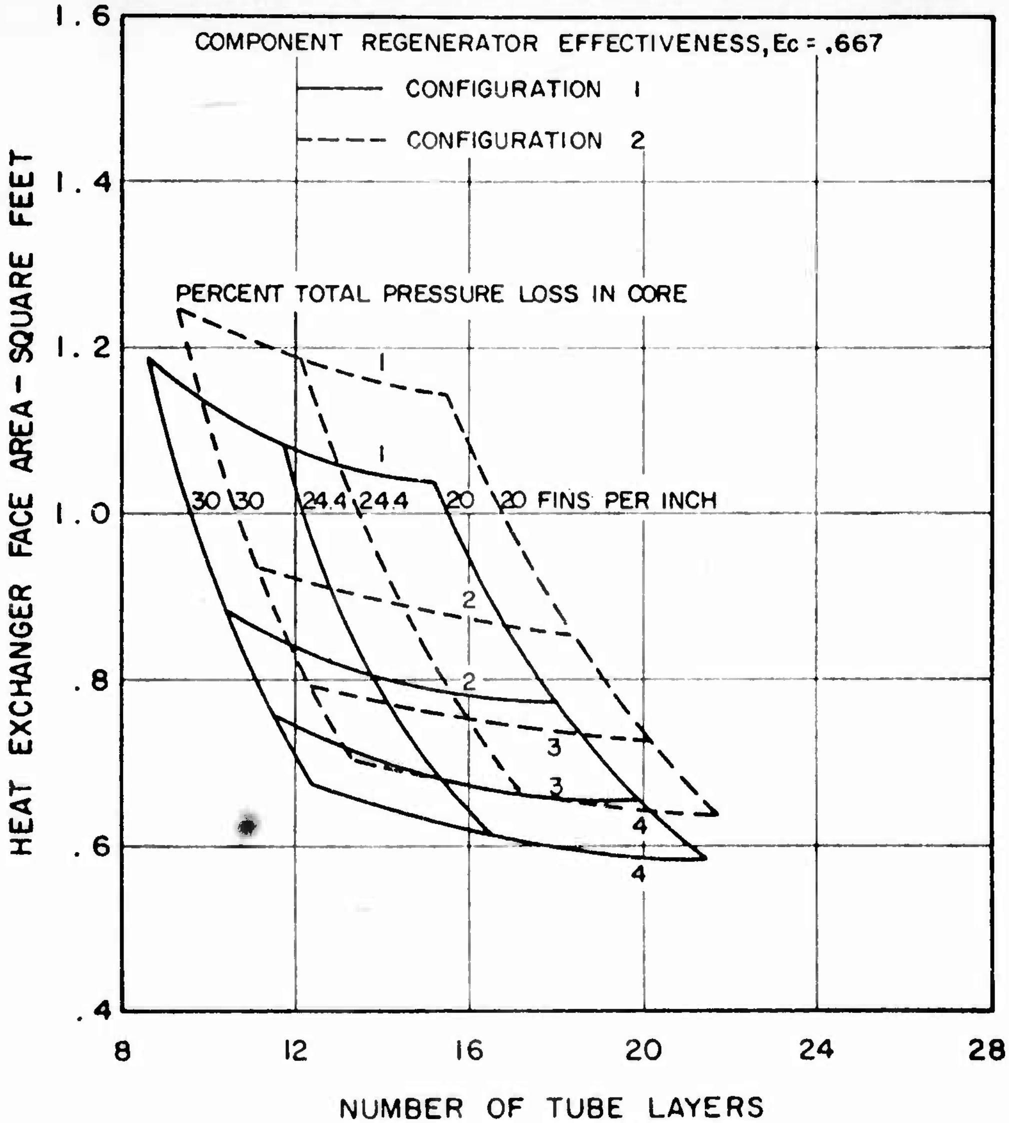


Figure d

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

FACE AREA vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND COMPRESSOR

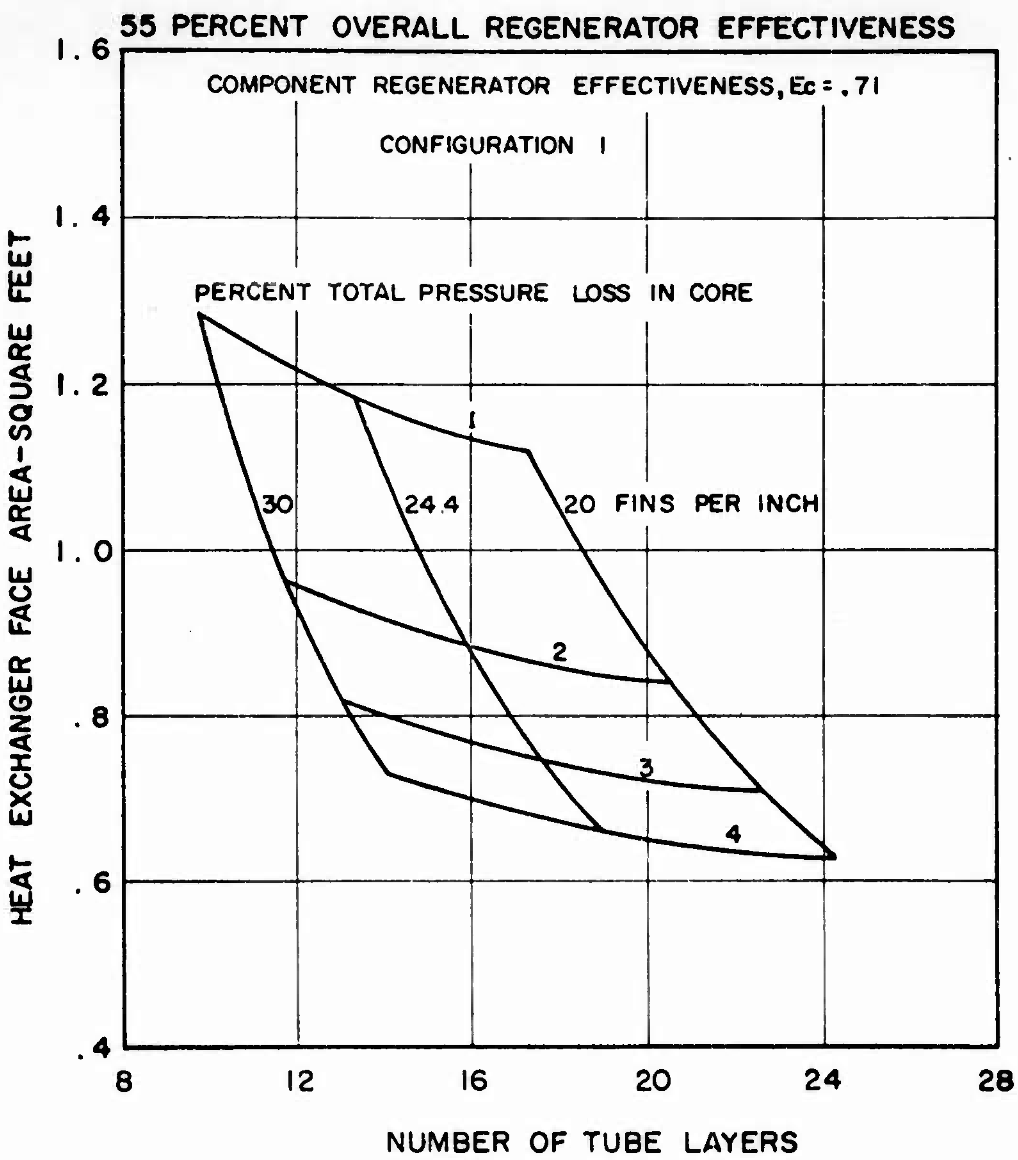


Figure e

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

FACE AREA vs NUMBER OF TUBE LAYERS

HEAT EXCHANGER BEHIND COMPRESSOR

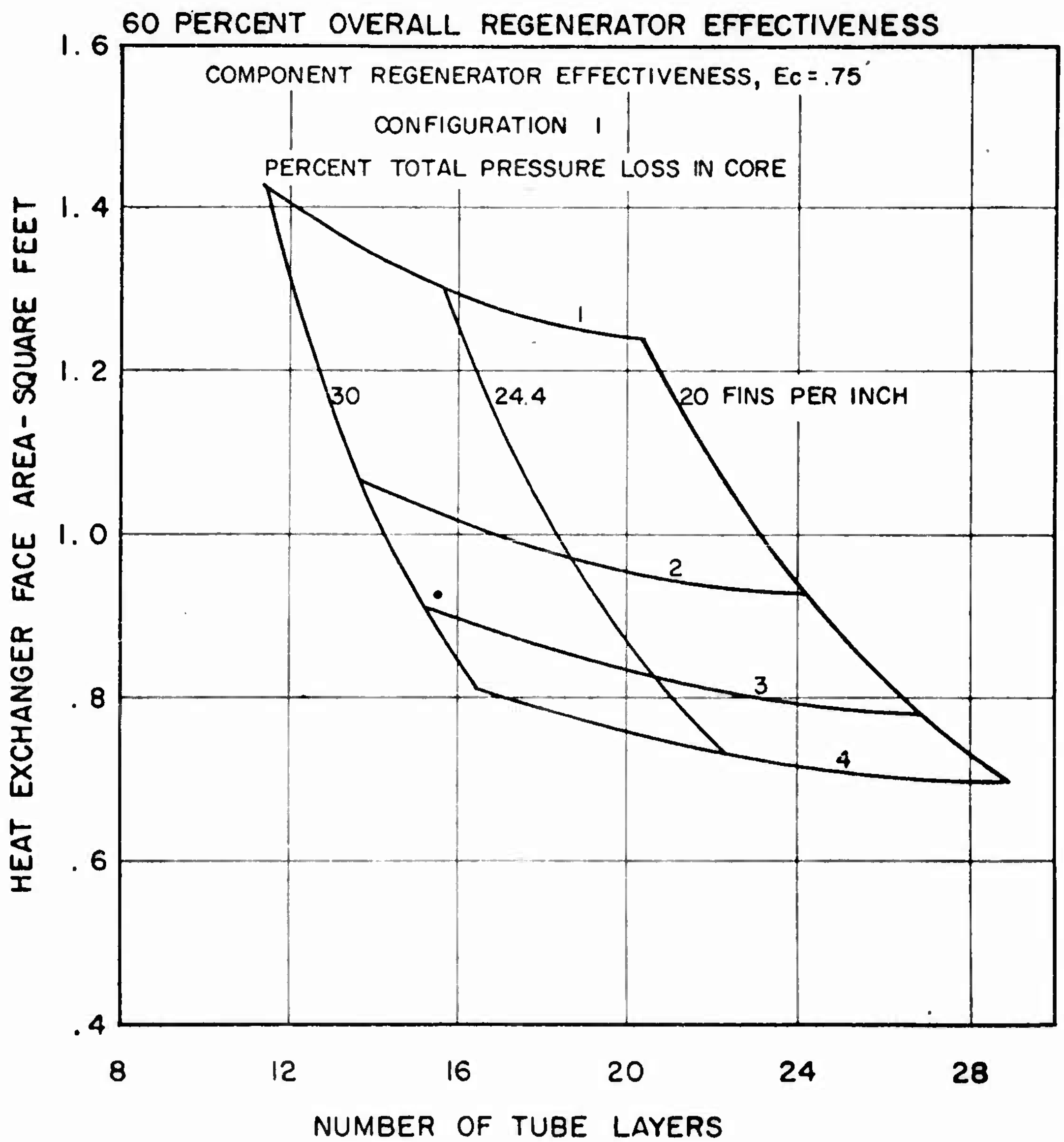


Figure f

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE WEIGHT vs NUMBER OF TUBE LAYERS

HEAT EXCHANGER BEHIND COMPRESSOR

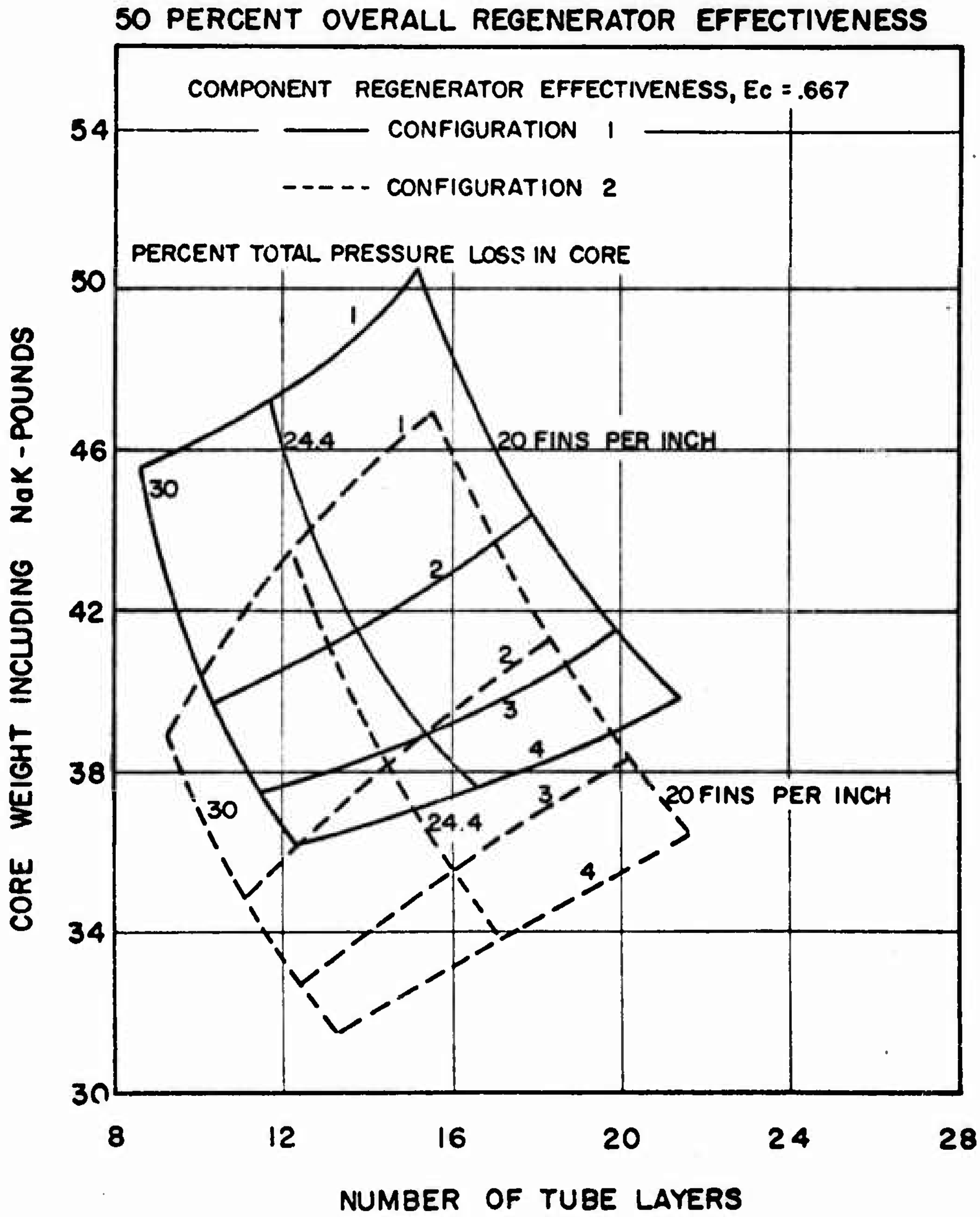


Figure 8

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE WEIGHT vs NUMBER OF TUBE LAYERS

HEAT EXCHANGER BEHIND COMPRESSOR

55 PERCENT OVERALL REGENERATOR EFFECTIVENESS

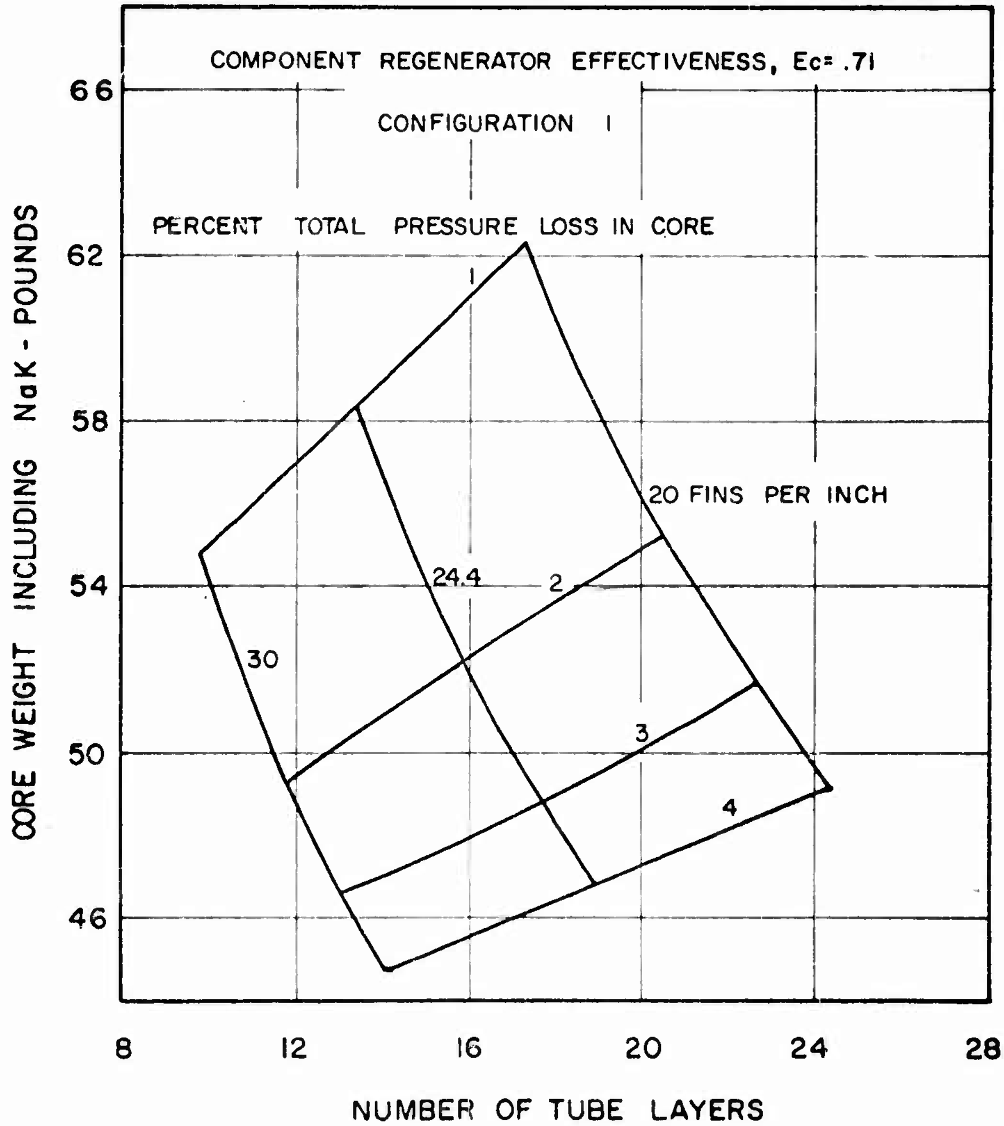


Figure h

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE WEIGHT vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND COMPRESSOR

60 PERCENT OVERALL REGENERATOR EFFECTIVENESS

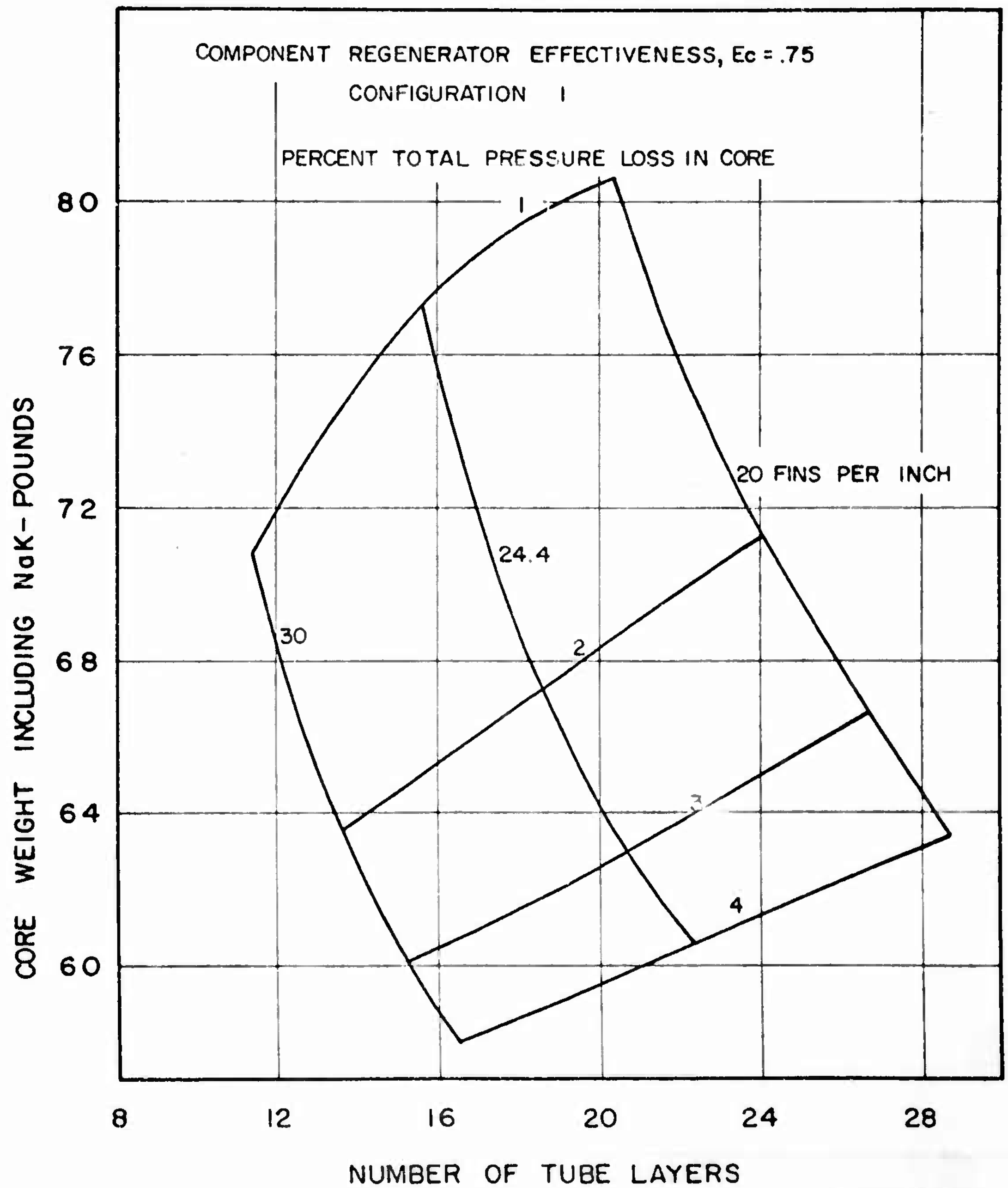


Figure 1

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE TOTAL PRESSURE LOSS vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND TURBINE

50 PERCENT OVERALL REGENERATOR EFFECTIVENESS

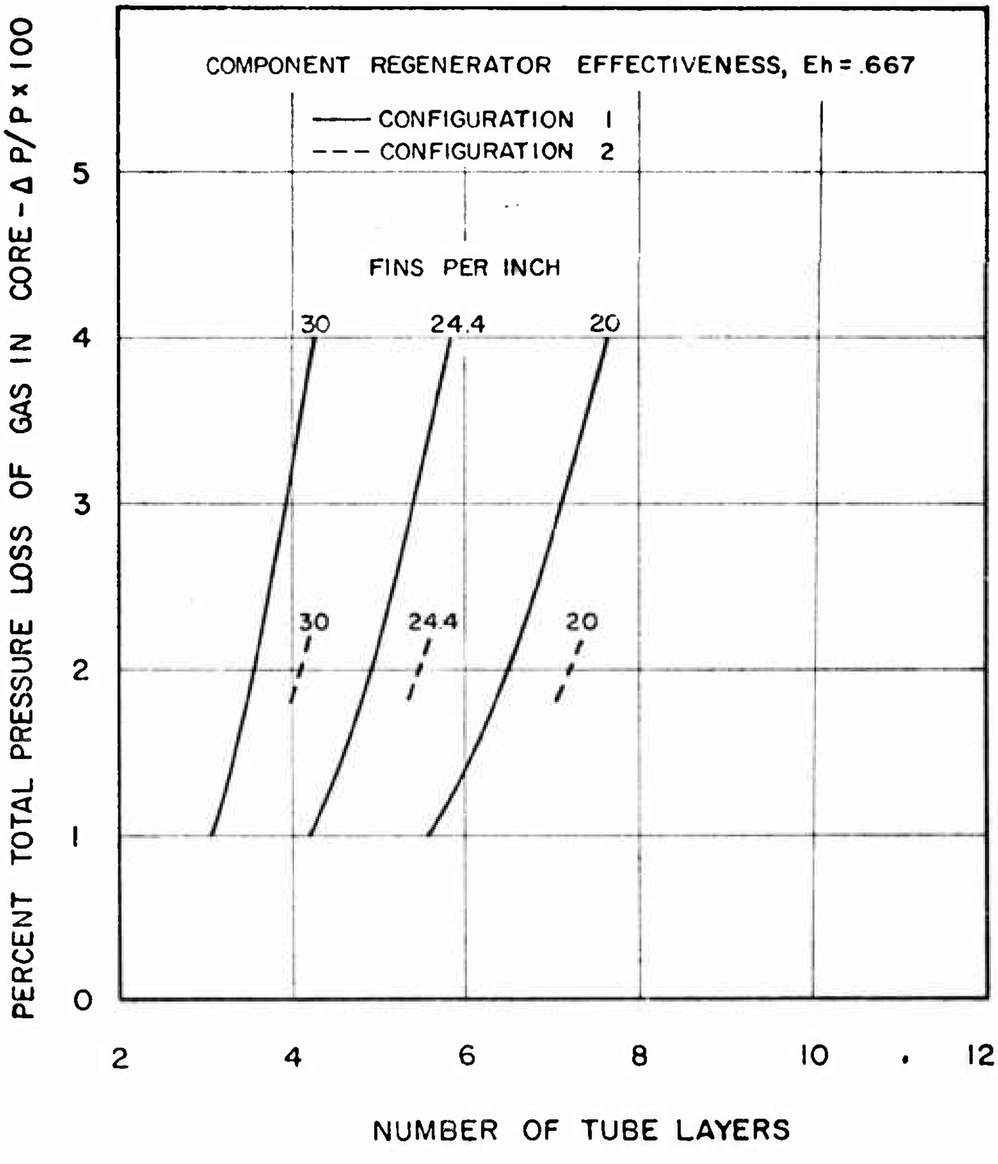


Figure j

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE TOTAL PRESSURE LOSS vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND TURBINE

55 PERCENT OVERALL REGENERATOR EFFECTIVENESS

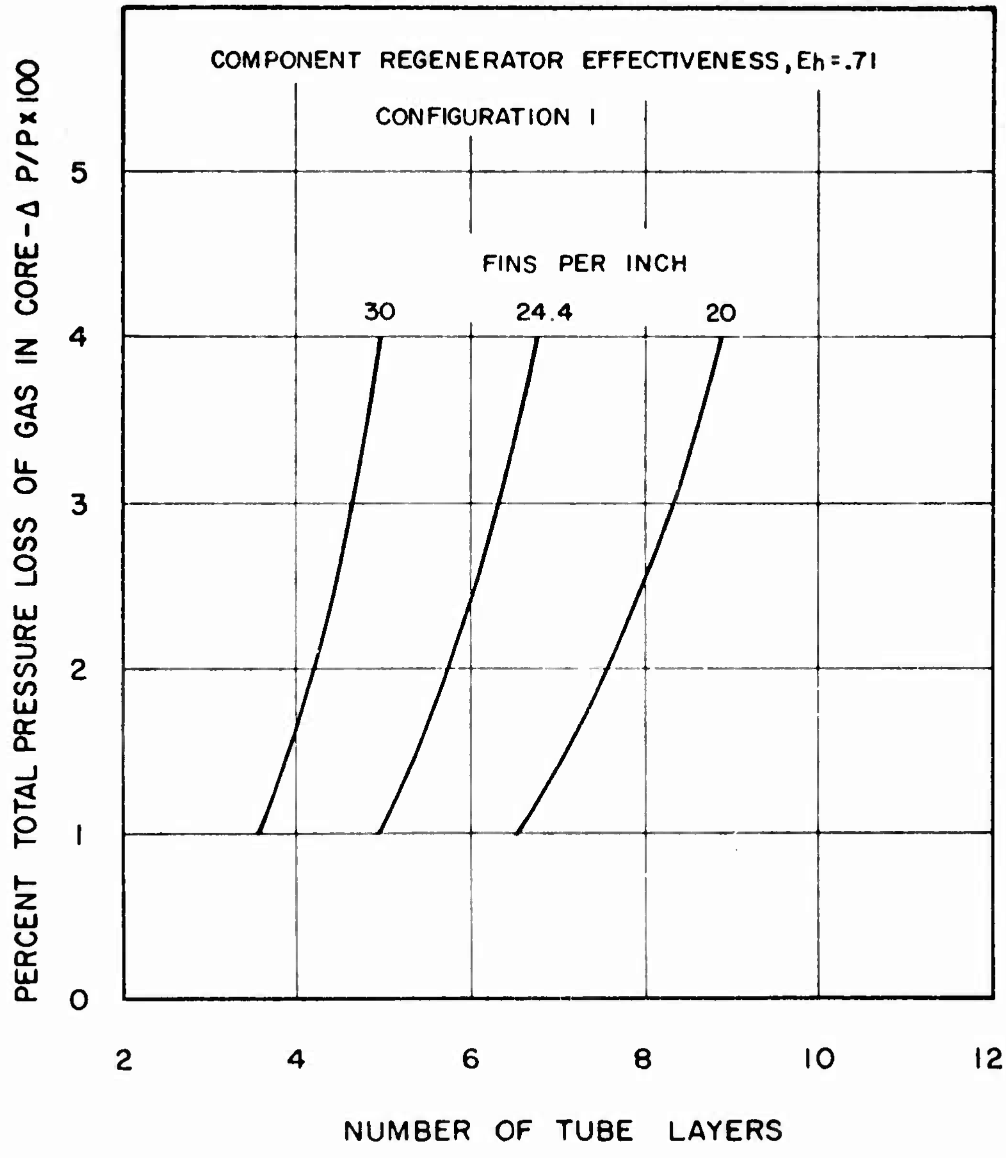


Figure k

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE TOTAL PRESSURE LOSS vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND TURBINE

60 PERCENT OVERALL REGENERATOR EFFECTIVENESS

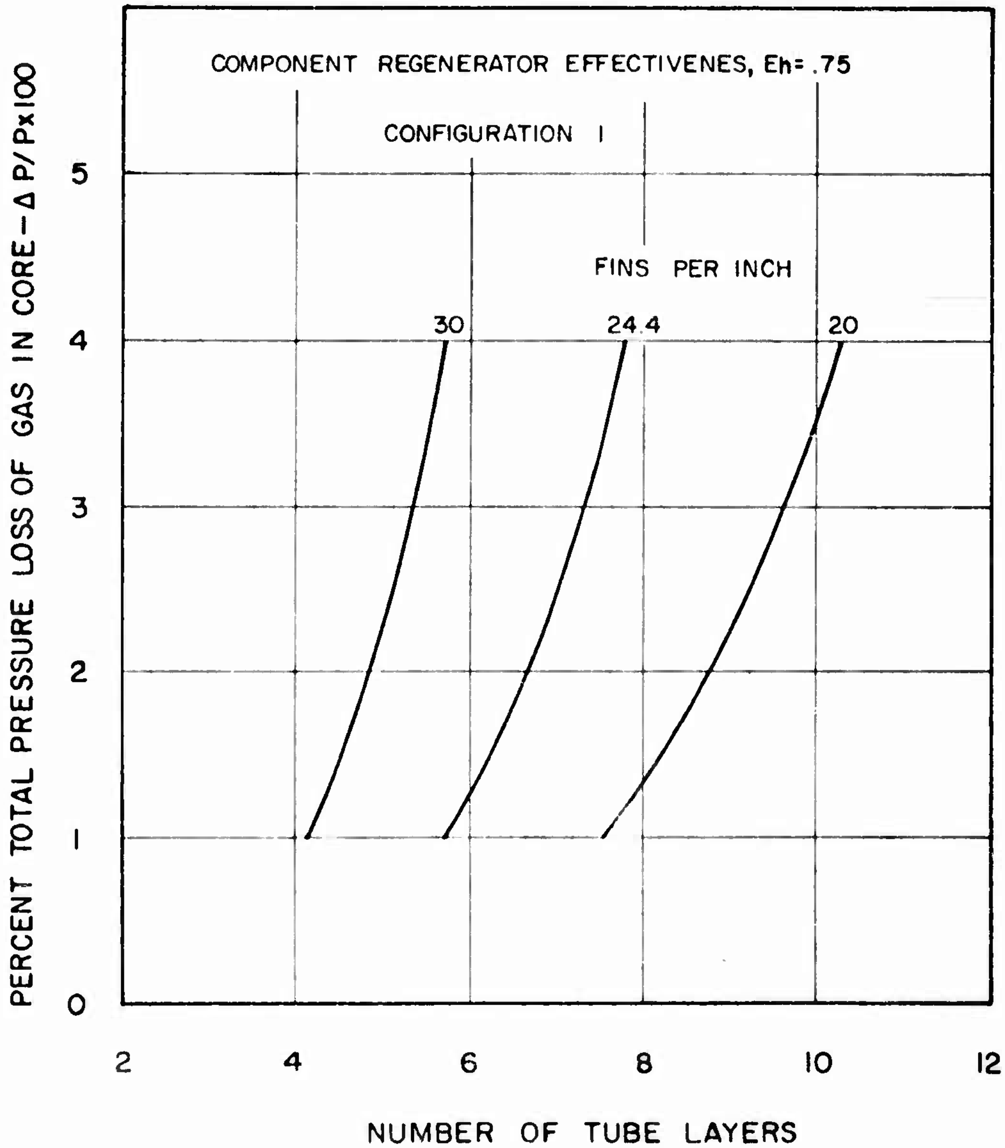


Figure 1

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

FACE AREA vs NUMBER OF TUBE LAYERS
HEAT EXCHANGE BEHIND TURBINE

50 PERCENT OVERALL REGENERATOR EFFECTIVENESS

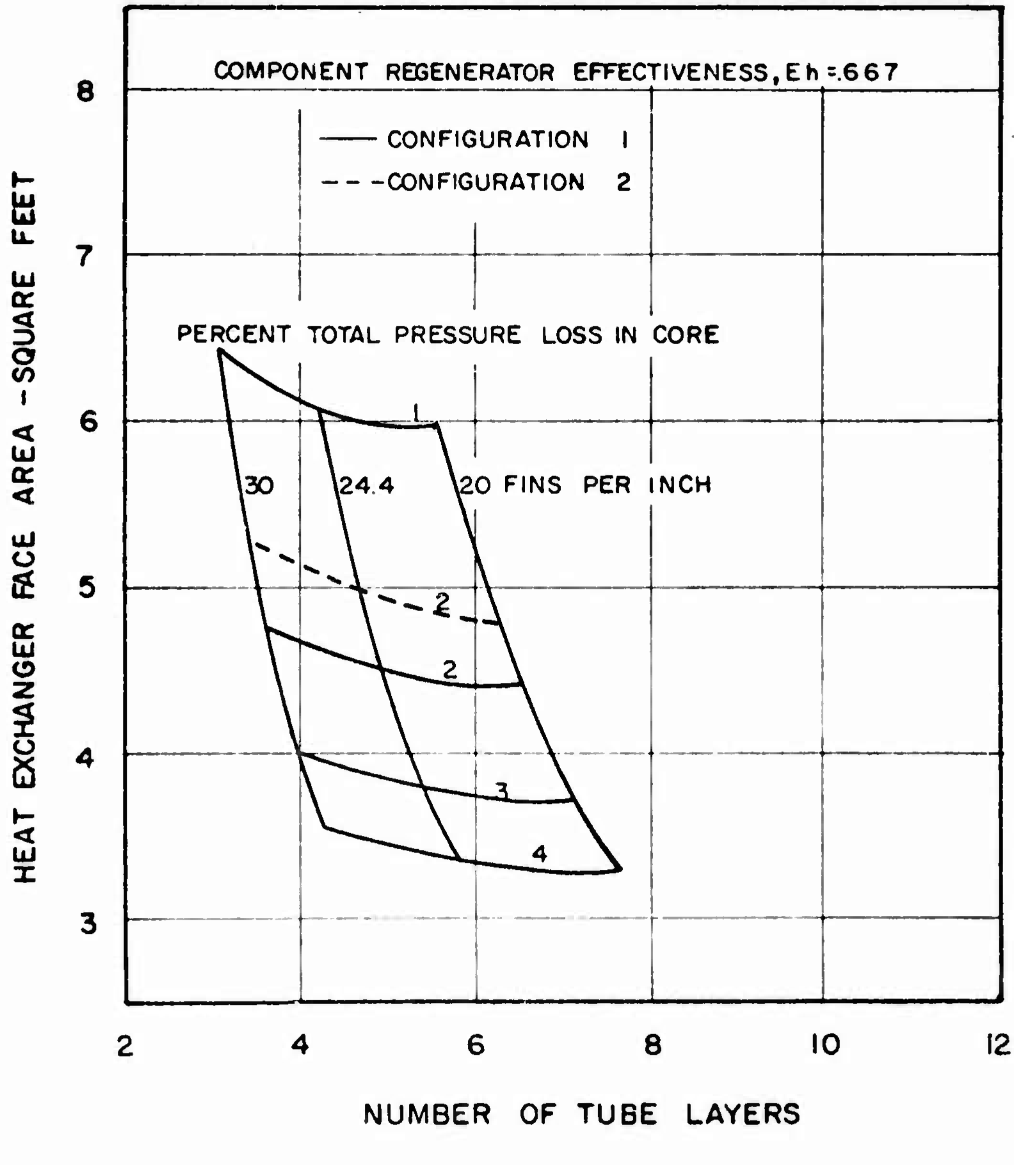


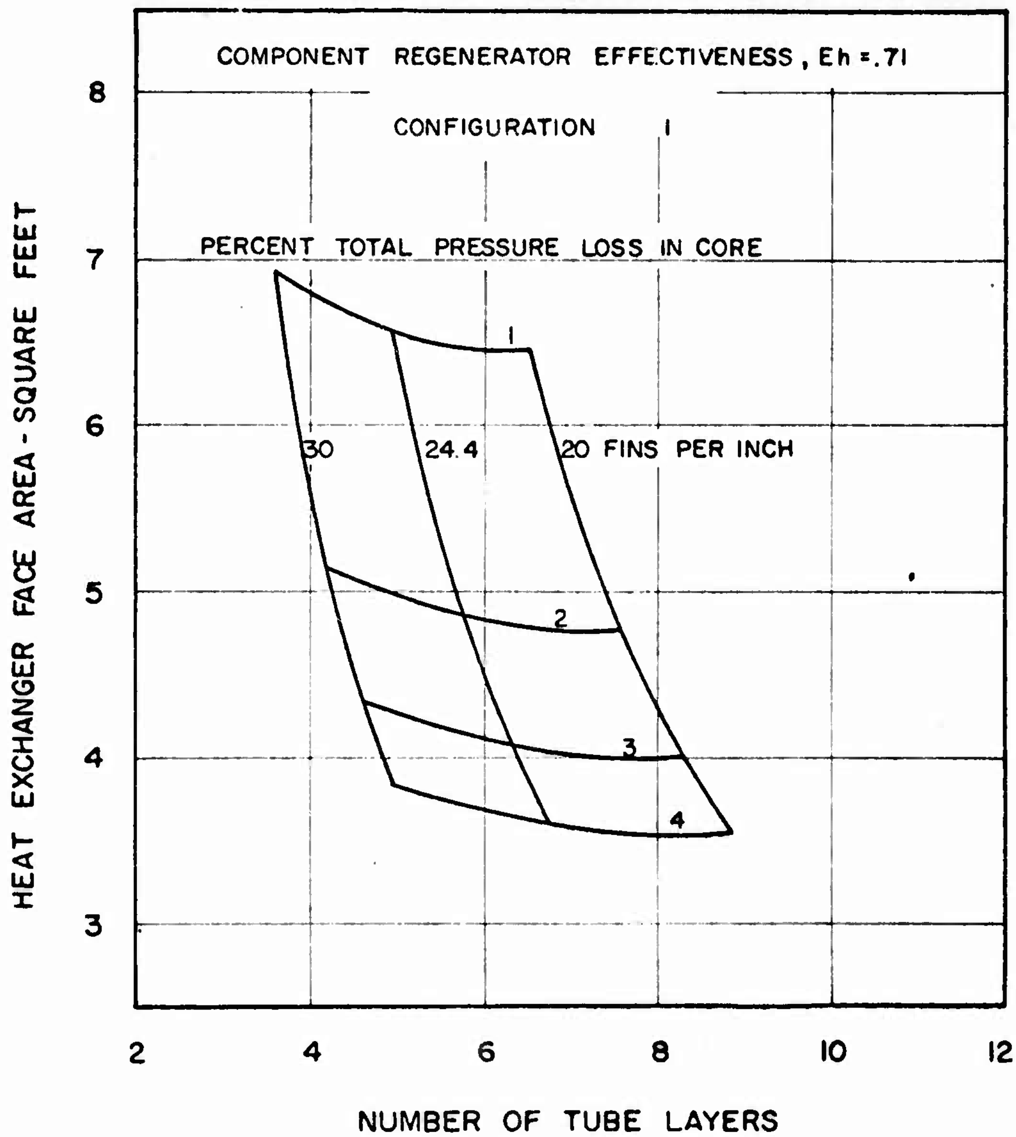
Figure m

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

FACE AREA vs NUMBER OF TUBE LAYERS

HEAT EXCHANGER BEHIND TURBINE

55 PERCENT OVERALL REGENERATOR EFFECTIVENESS



LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE
FACE AREA vs NUMBER OF TUBE LAYERS
HEAT EXCHANGE BEHIND TURBINE

60 PERCENT OVERALL REGENERATOR EFFECTIVENESS

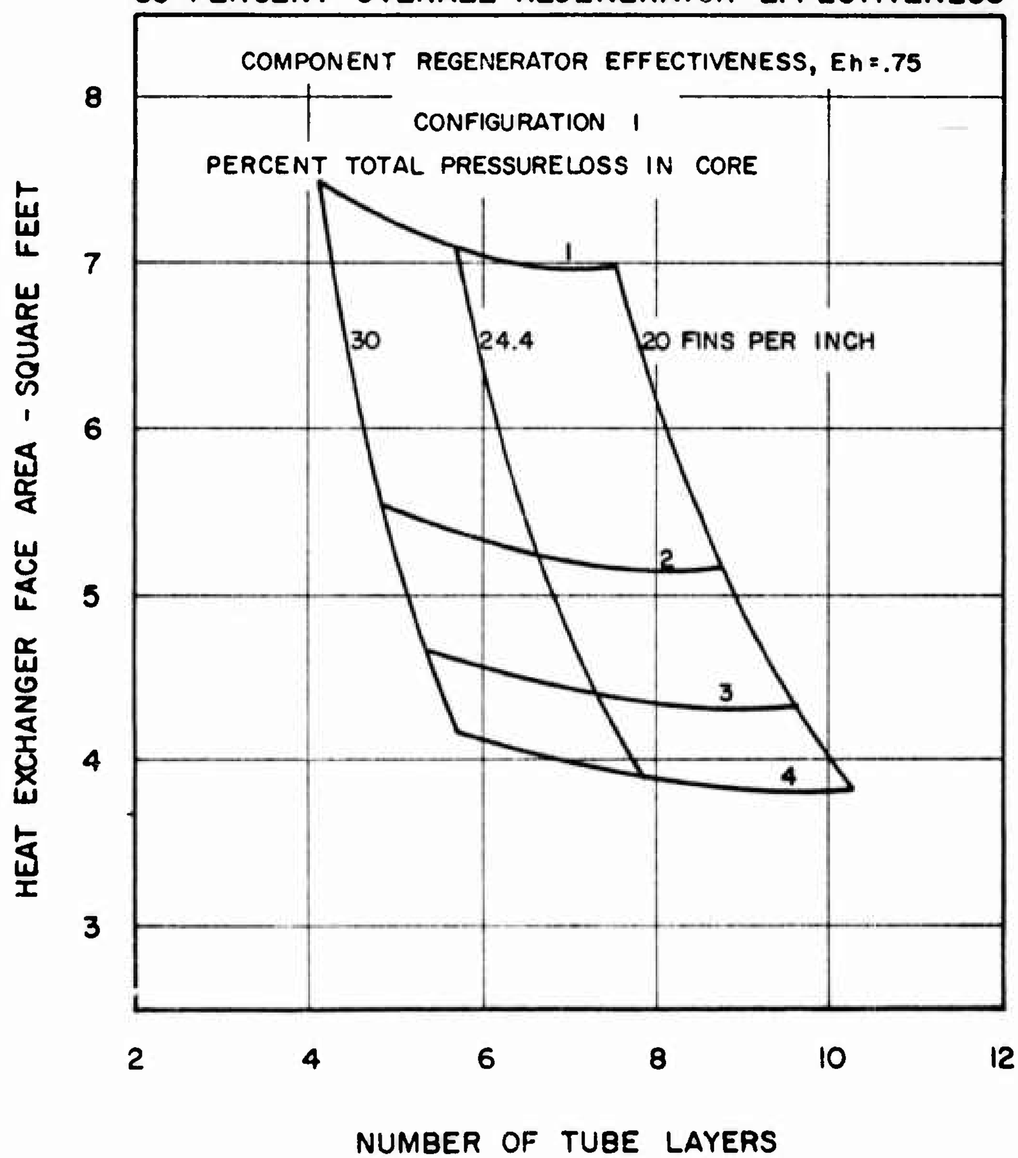


Figure 6

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE WEIGHT vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND TURBINE

50 PERCENT OVERALL REGENERATOR EFFECTIVENESS

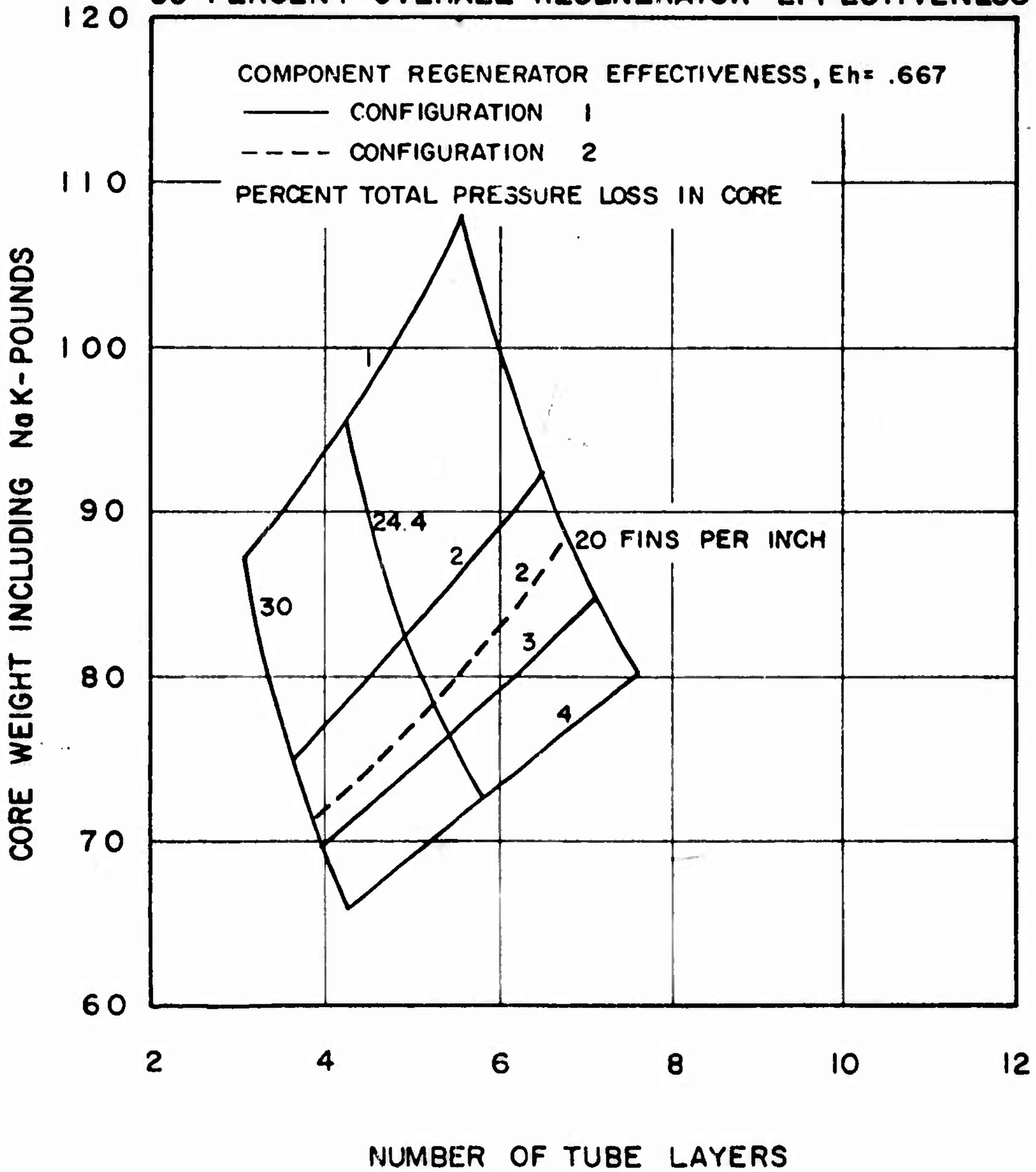


Figure p

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE WEIGHT vs NUMBER OF TUBE LAYERS
HEAT EXCHANGER BEHIND TURBINE

55 PERCENT OVERALL REGENERATOR EFFECTIVENESS

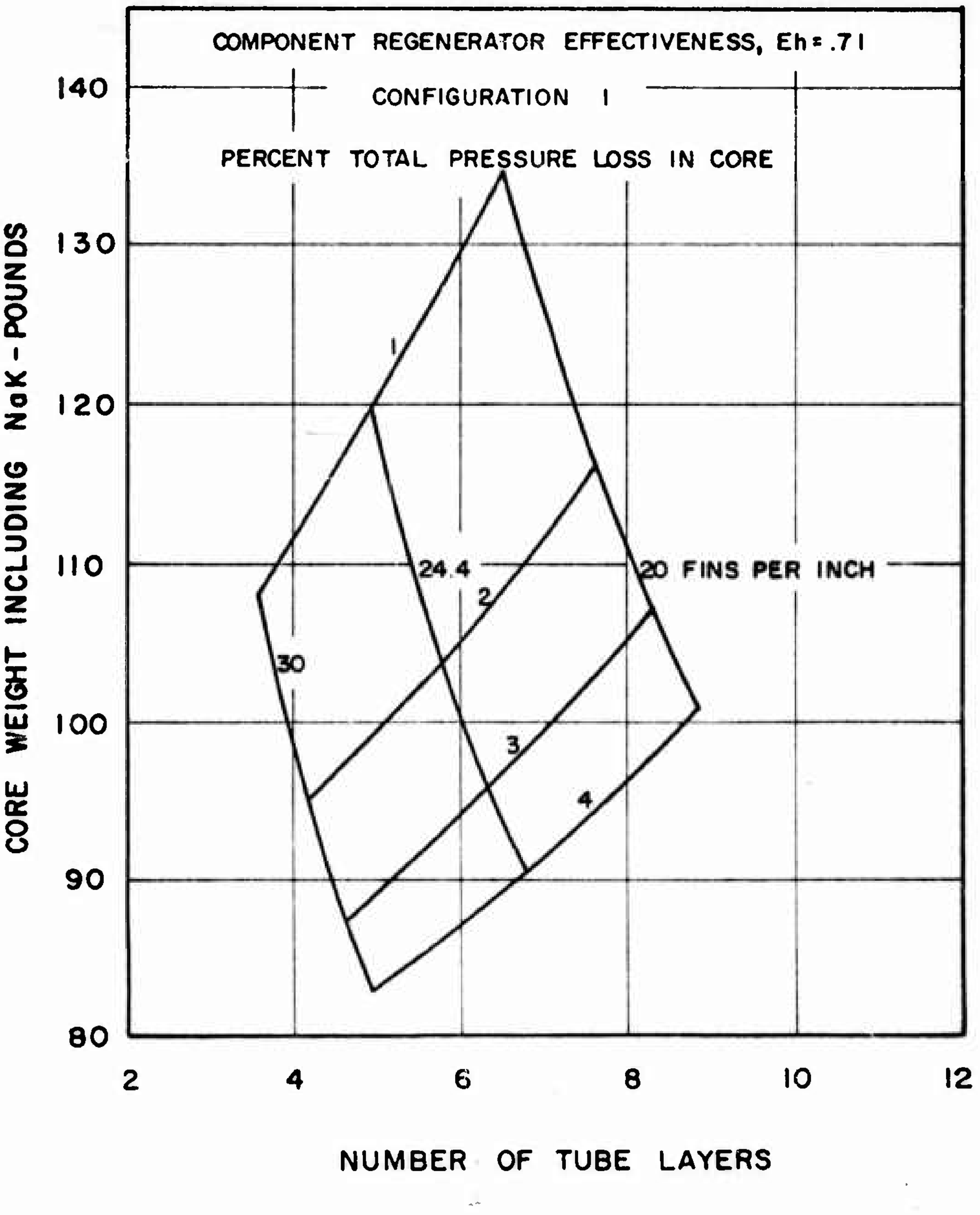


Figure 9

LIQUID METAL REGENERATOR FEASIBILITY STUDY

1000 HORSEPOWER TURBOSHAFT ENGINE

CORE WEIGHT vs NUMBER OF TUBE LAYERS

HEAT EXCHANGER BEHIND TURBINE

60 PERCENT OVERALL REGENERATOR EFFECTIVENESS

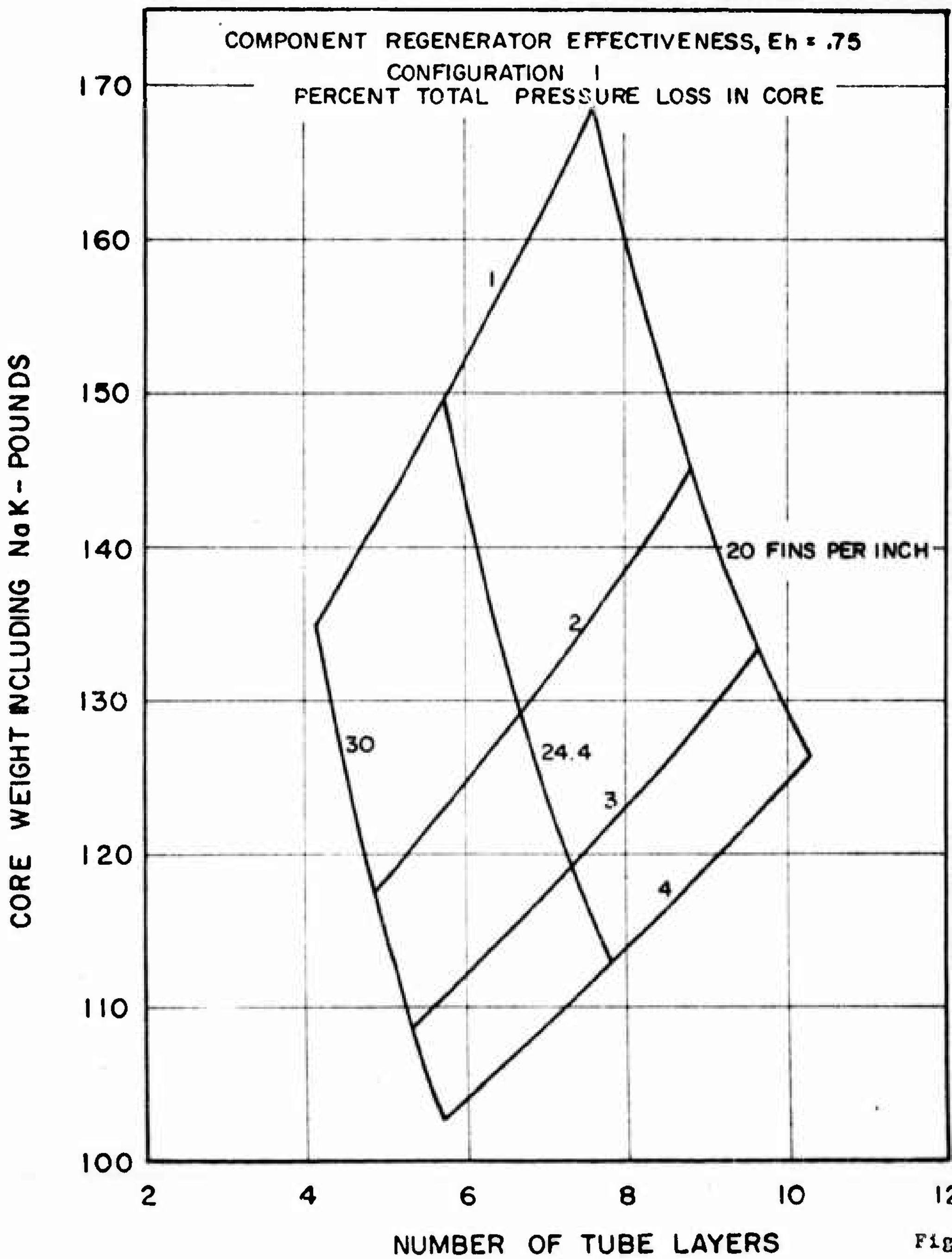


Figure r

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

TOTAL CORE WEIGHT OF BOTH EXCHANGERS
vs
OVERALL REGENERATOR EFFECTIVENESS

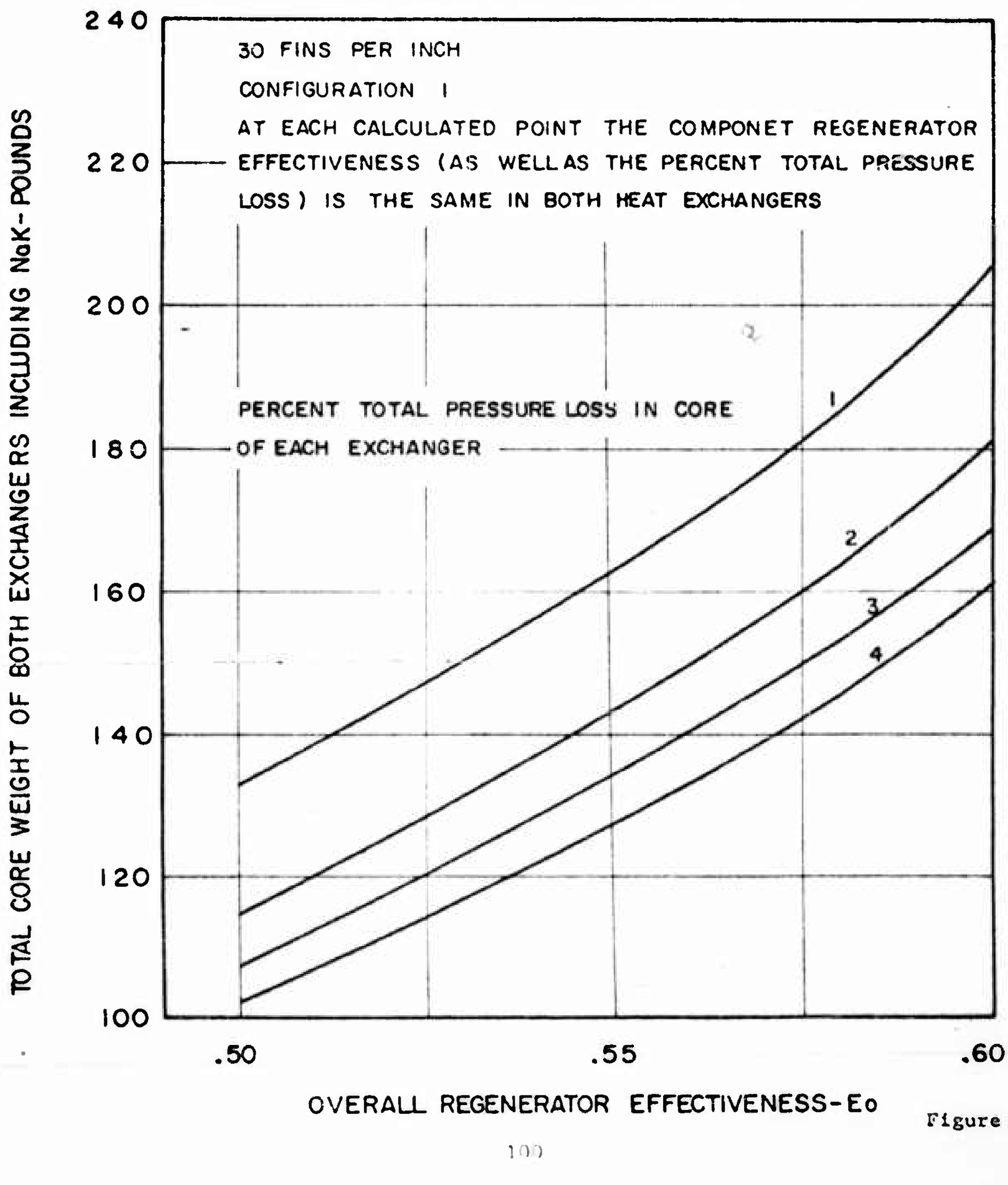


Figure 8

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

TOTAL CORE WEIGHT OF BOTH EXCHANGERS
VS
OVERALL REGENERATOR EFFECTIVENESS

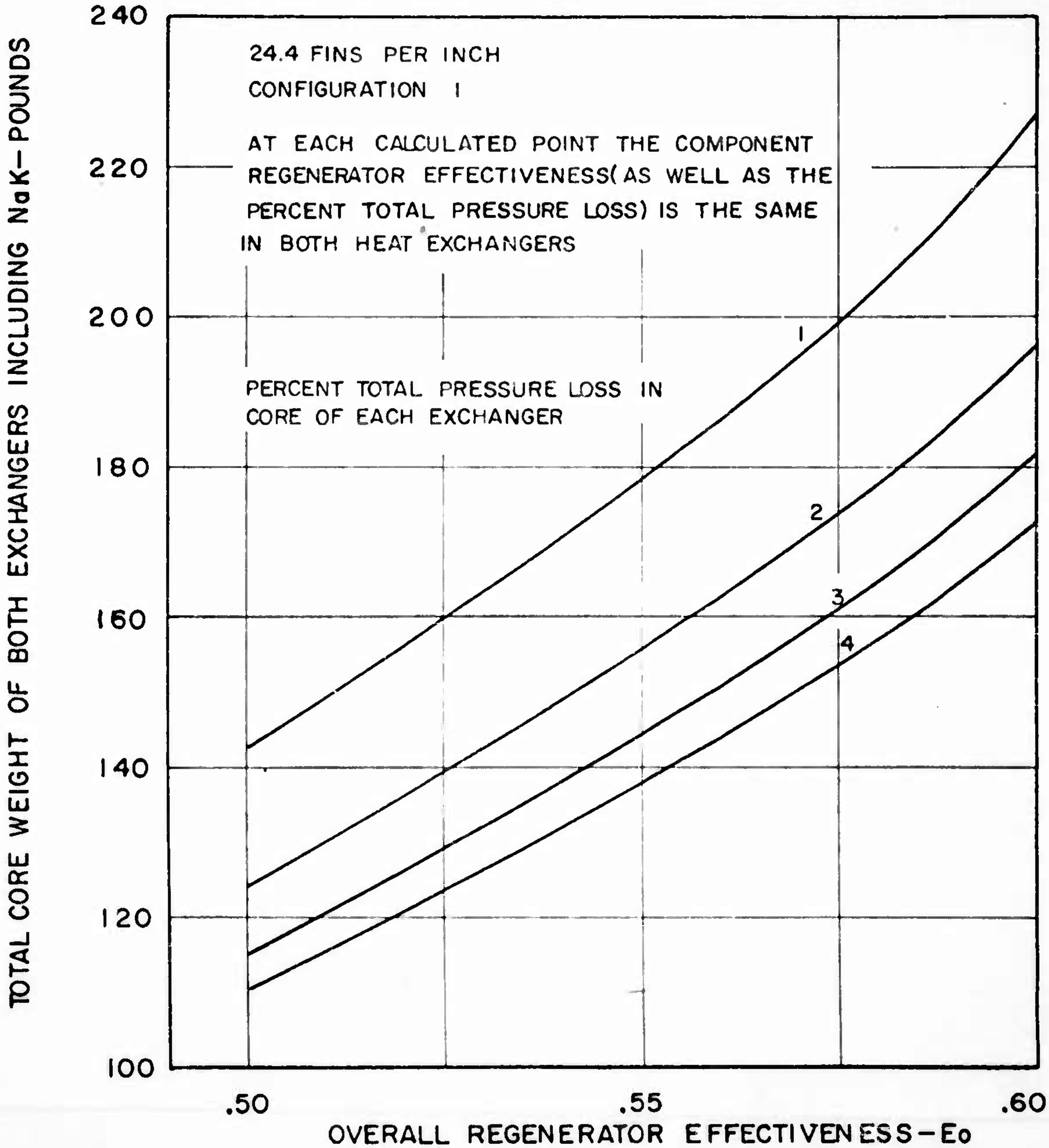


Figure t

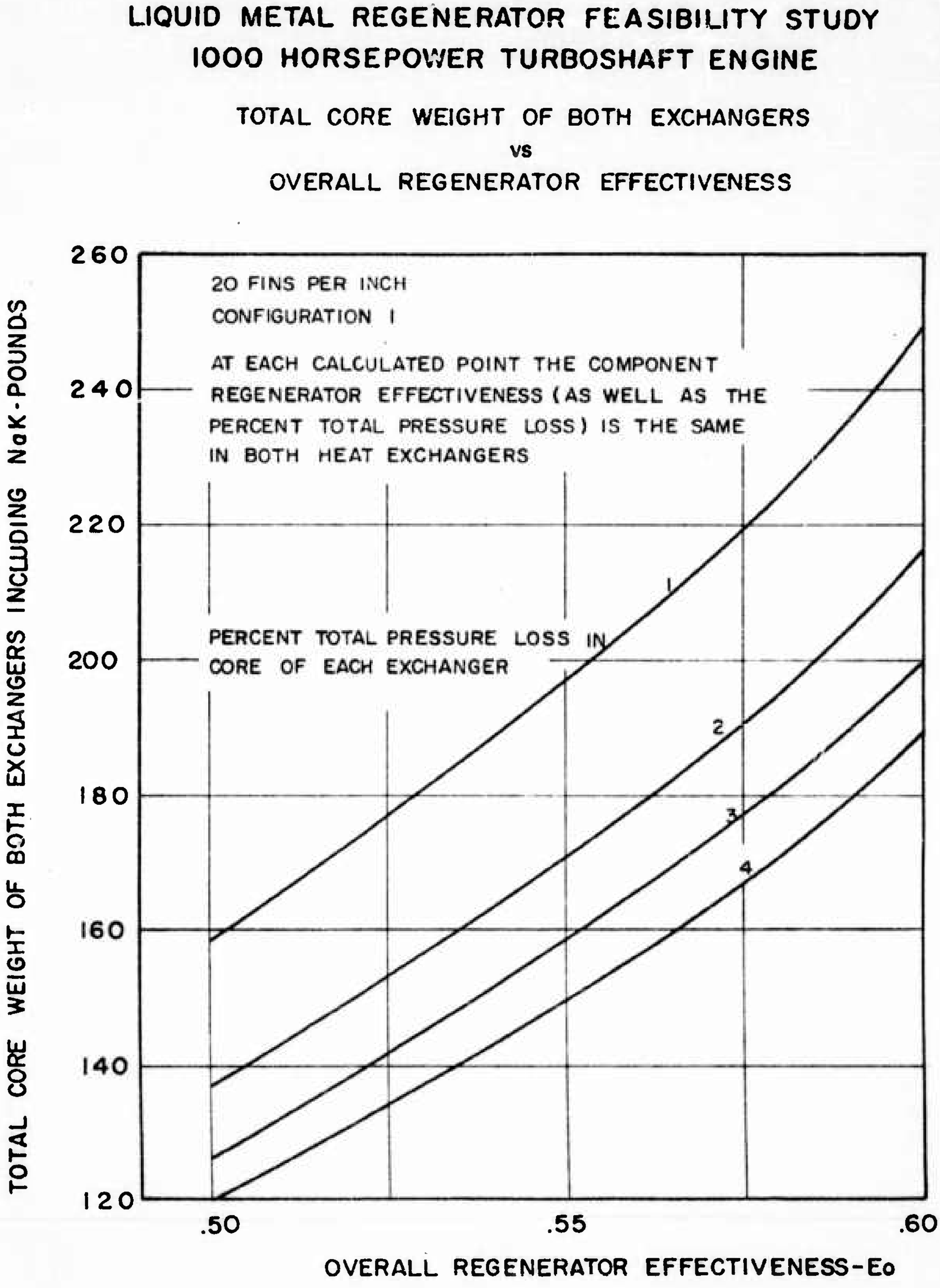

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE
TOTAL CORE WEIGHT OF BOTH EXCHANGERS
VS
OVERALL REGENERATOR EFFECTIVENESS
20 FINS PER INCH
CONFIGURATION I
AT EACH CALCULATED POINT THE COMPONENT REGENERATOR EFFECTIVENESS (AS WELL AS THE PERCENT TOTAL PRESSURE LOSS) IS THE SAME IN BOTH HEAT EXCHANGERS
PERCENT TOTAL PRESSURE LOSS IN CORE OF EACH EXCHANGER
1
2
3
4
260
240
220
200
180
160
140
120
.50
.55
.60
TOTAL CORE WEIGHT OF BOTH EXCHANGERS INCLUDING NaK-POUNDS
OVERALL REGENERATOR EFFECTIVENESS-Eo

Figure u

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

TOTAL CORE WEIGHT OF BOTH EXCHANGERS
vs
NUMBER OF FINS PER INCH

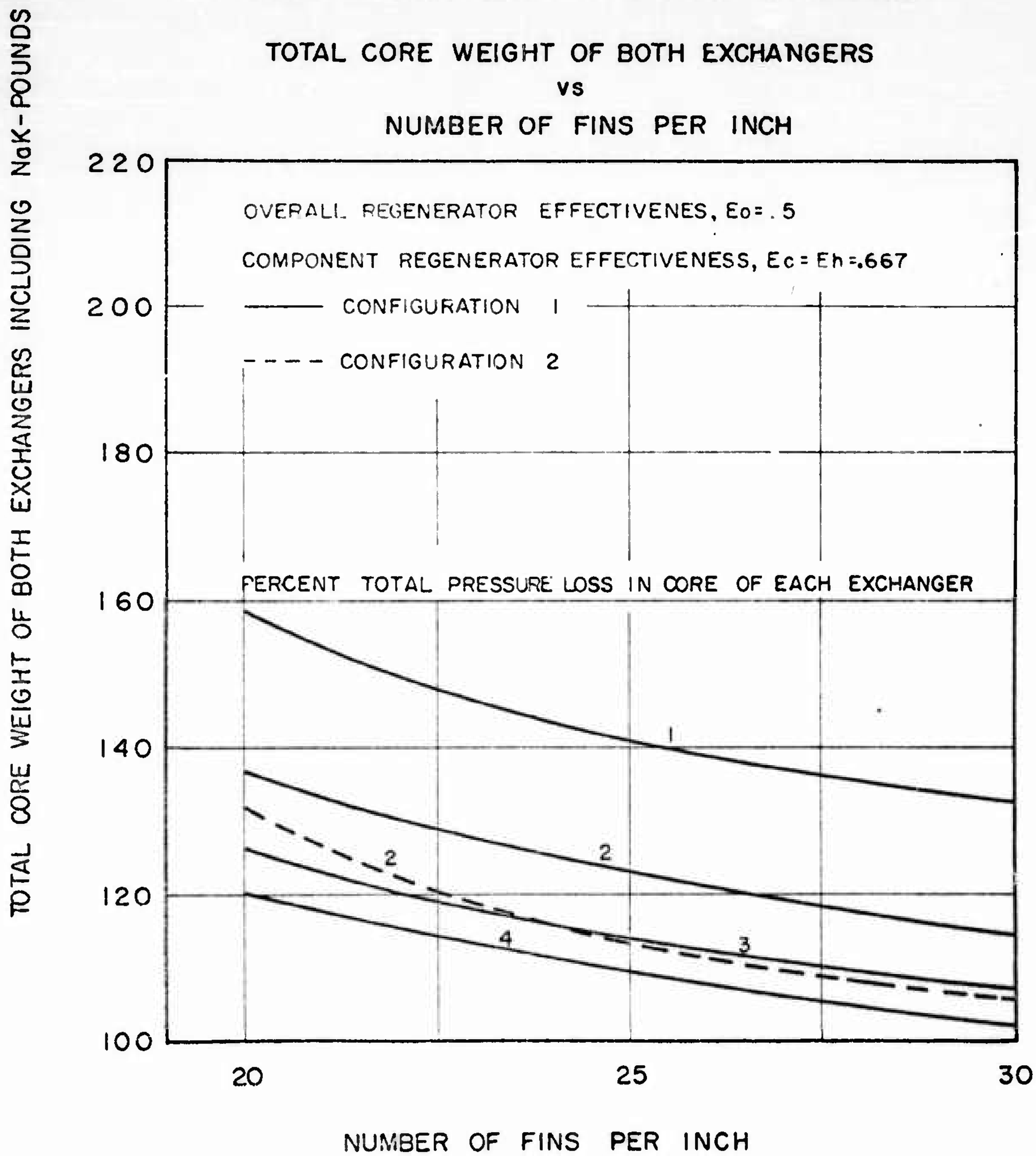


Figure v

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

TOTAL CORE WEIGHT OF BOTH EXCHANGERS
VS
NUMBER OF FINS PER INCH

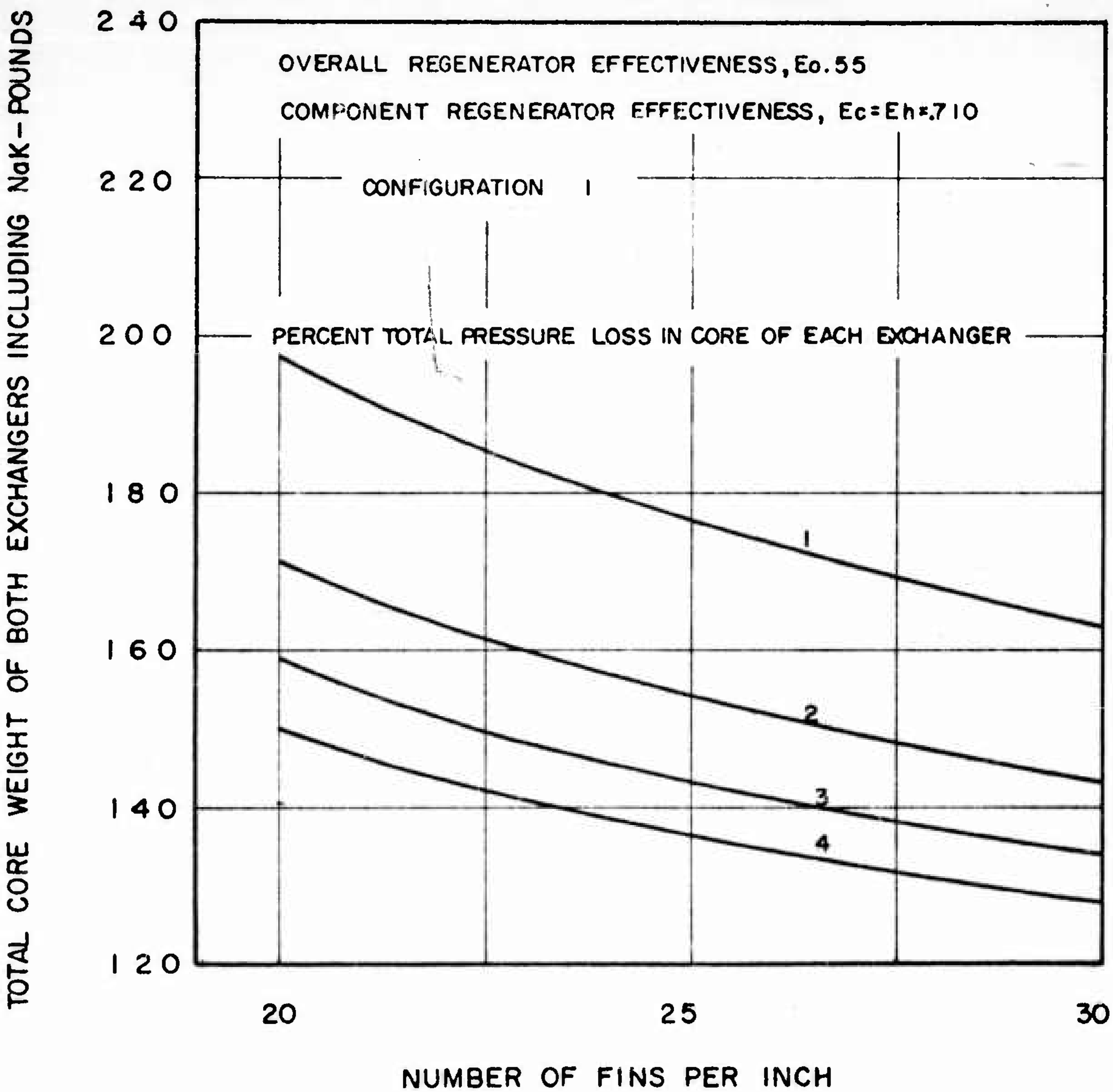


Figure w

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

TOTAL CORE WEIGHT OF BOTH EXCHANGERS
VS
NUMBER OF FINS PER INCH

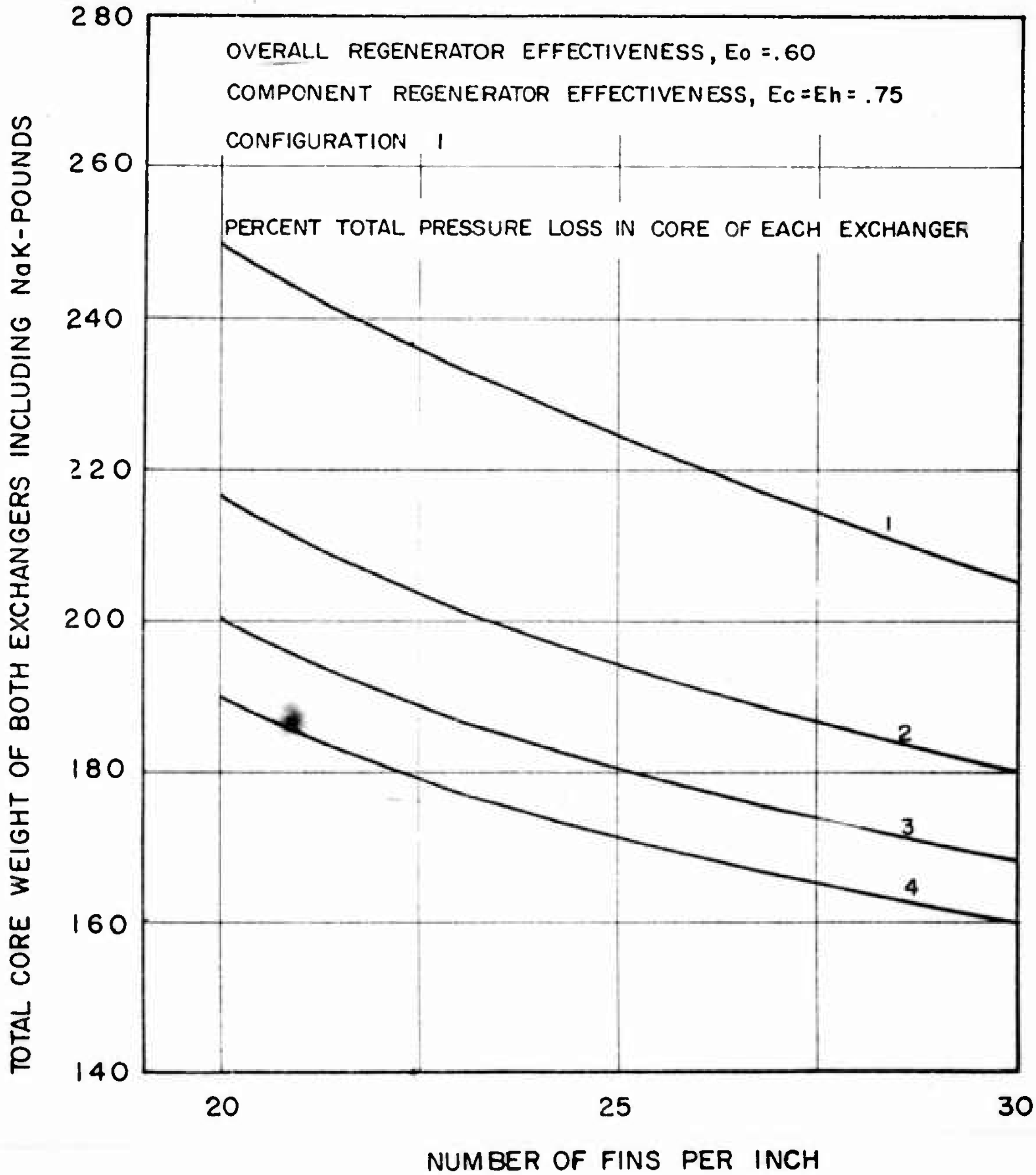


Figure x

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

CORE WEIGHTS vs FIN THICKNESS

FOR HEAT EXCHANGERS USING STAINLESS STEEL CLAD COPPER FINS AND COMPARED TO ALUMINUM FINS IN HEAT EXCHANGERS BEHIND COMPRESSOR.

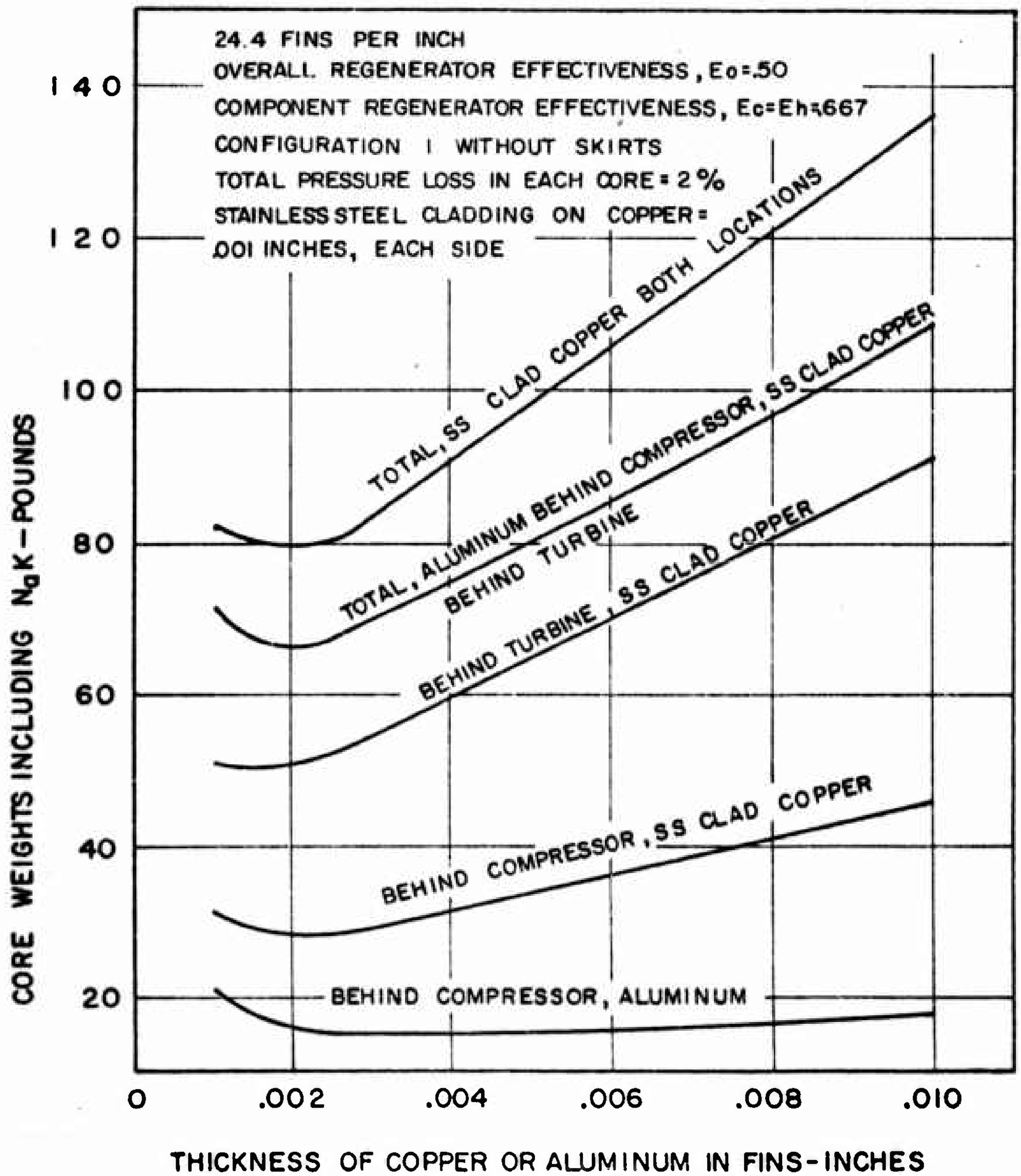


Figure y

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

NUMBER OF TUBE LAYERS vs FIN THICKNESS

FOR HEAT EXCHANGERS USING STAINLESS STEEL CLAD COPPER FINS AND COMPARED TO ALUMINUM FINS IN HEAT EXCHANGERS BEHIND COMPRESSOR

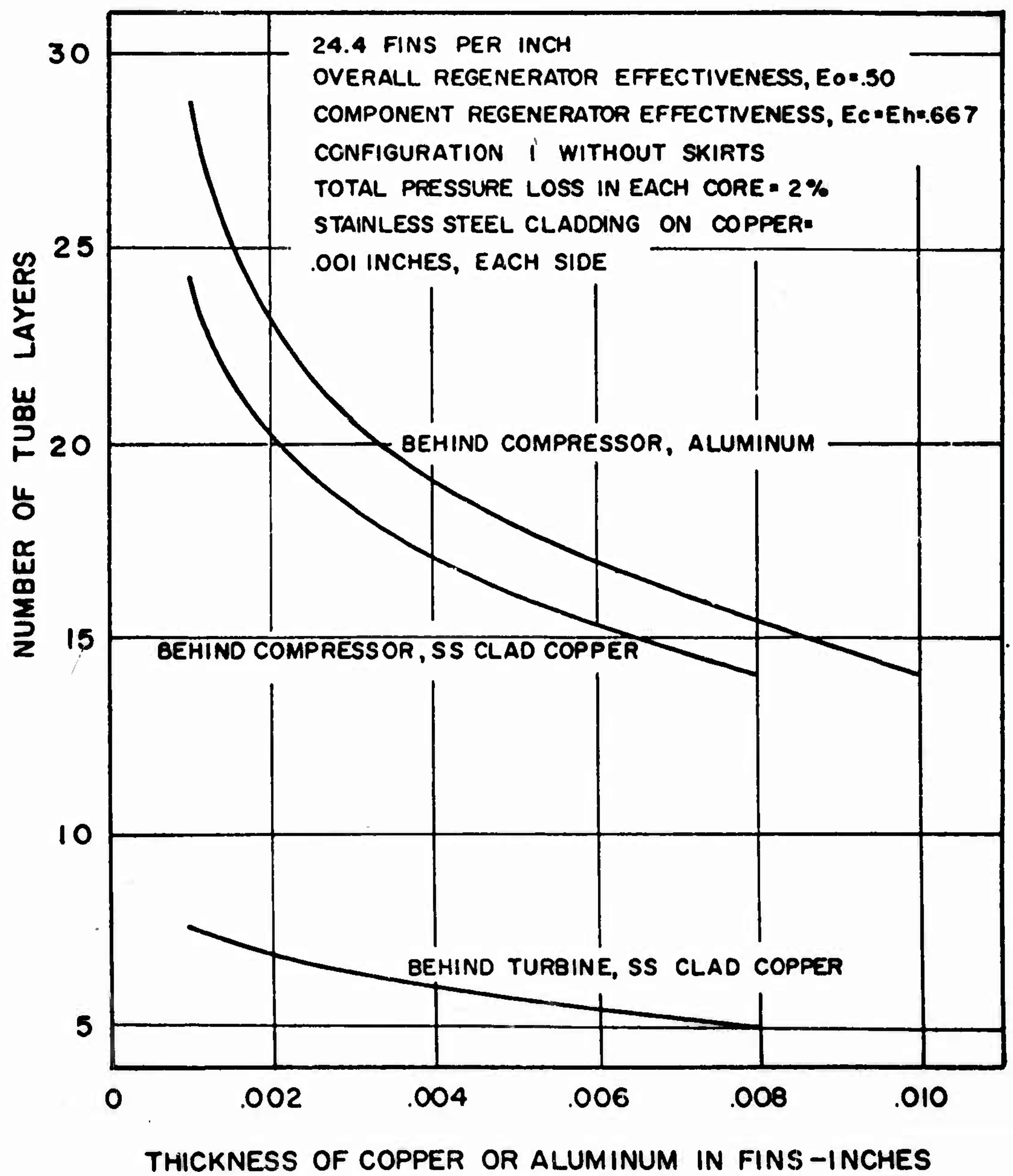


Figure z

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

FACE AREA vs FIN THICKNESS

HEAT EXCHANGER BEHIND TURBINE USING
STAINLESS STEEL CLAD COPPER FINS

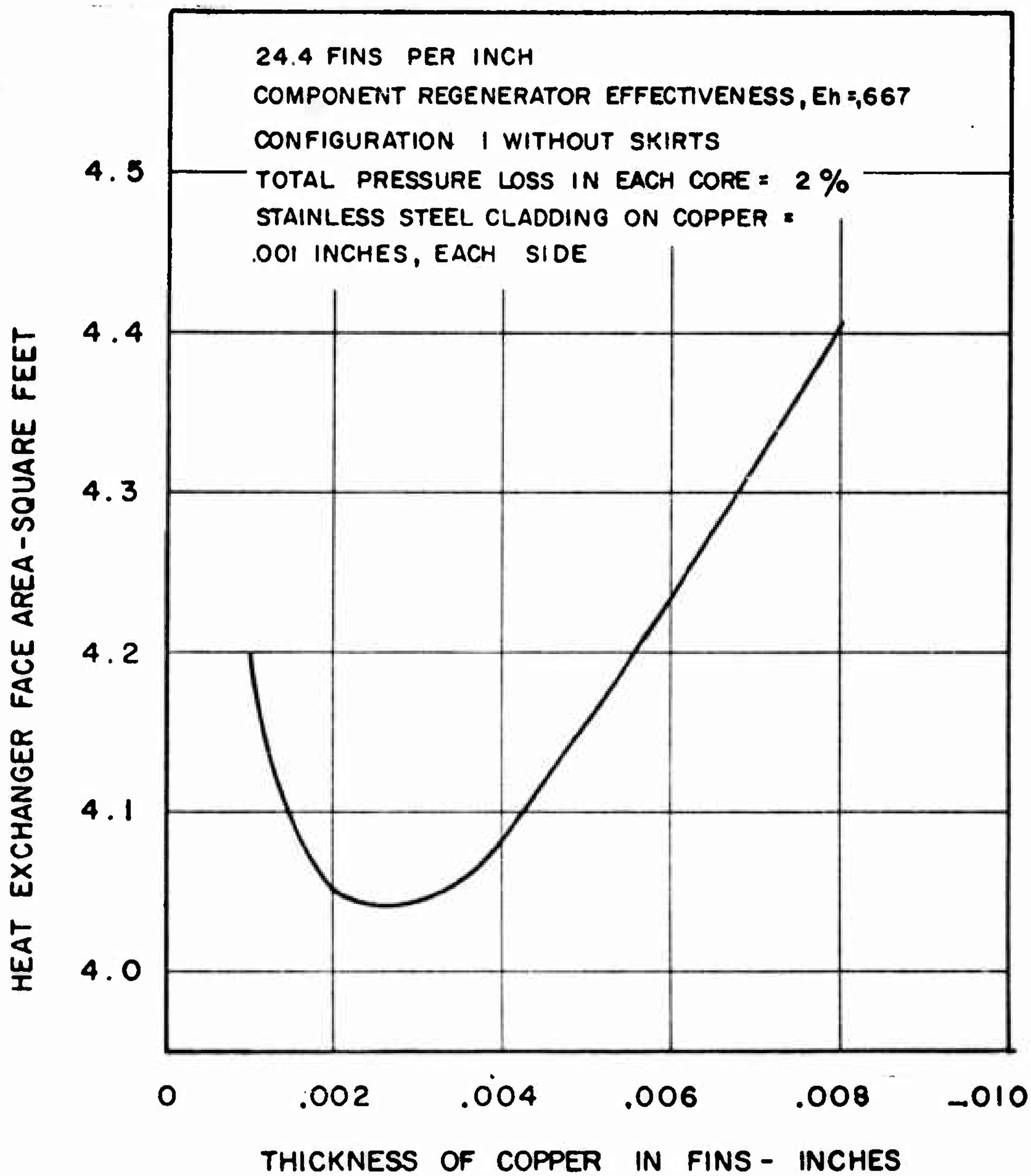


Figure aa

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

FACE AREA vs FIN THICKNESS

HEAT EXCHANGER BEHIND COMPRESSOR USING
STAINLESS STEEL CLAD COPPER AND ALUMINUM FINS

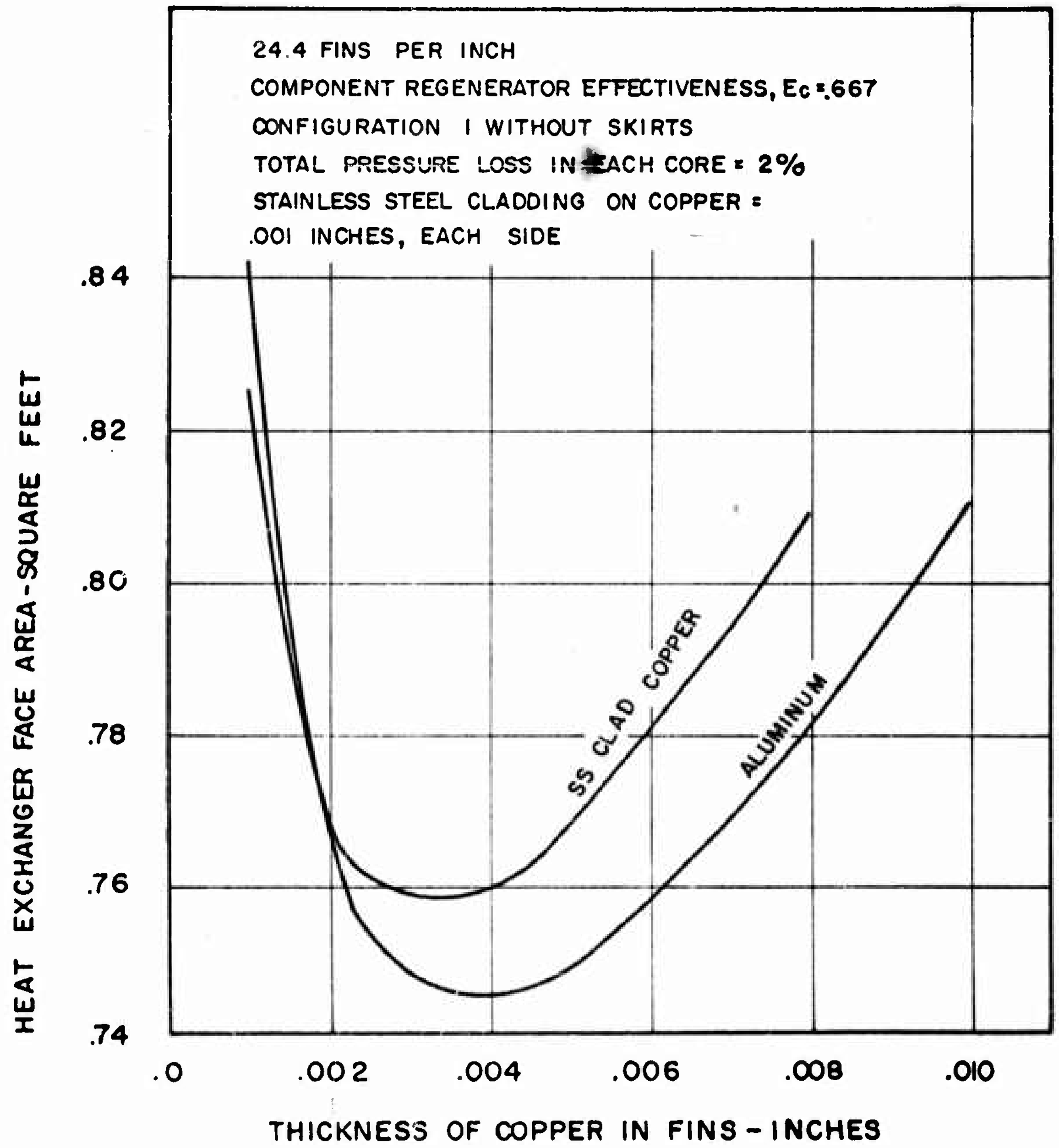


Figure bb

LIQUID METAL REGENERATOR FEASIBILITY STUDY
1000 HORSEPOWER TURBOSHAFT ENGINE

TOTAL CORE WEIGHT OF BOTH EXCHANGERS
vs
COMPONENT REGENERATOR EFFECTIVENESS RATIO

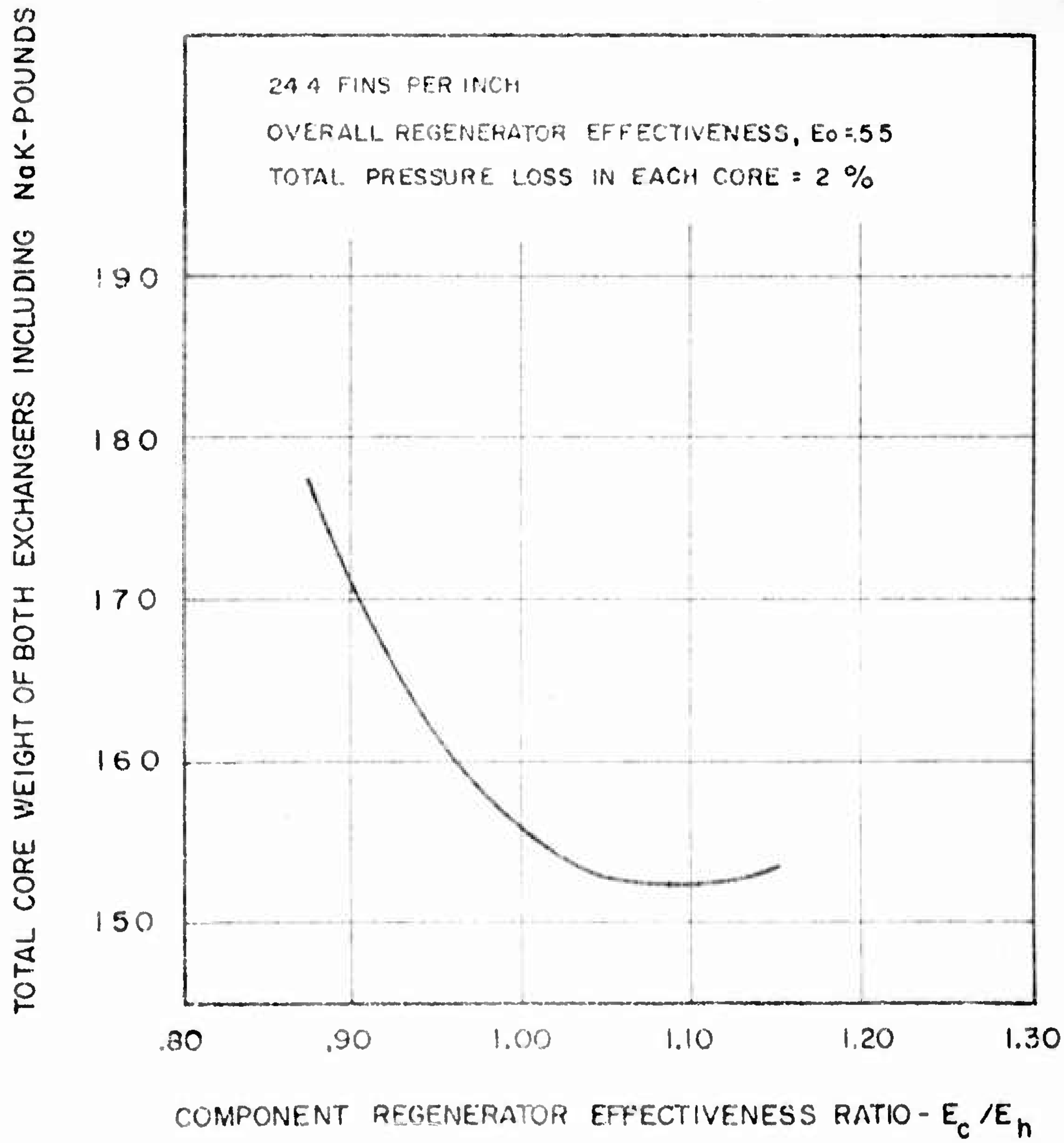


Figure cc

DISTRIBUTION

LIQUID METAL REGENERATOR FEASIBILITY STUDY
FOR
A LIGHTWEIGHT TURBOSHAFT ENGINE

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Curtiss-Wright Corp., Research Division, Quehanna, Pa.
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